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REVIEW OF ENGINE/AIRFRAME/DRIVE TRAIN DYNAMIC INTERFACE DEVELOPMENT PROBLEMS

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APPLIED TECHNOLOGY LABORATORY POSITION STATEMENT

This report provides the details of a program that is part of a larger effort designed to provide a complete report of past and present engine/airframe/drive train dynamic interface problems. The problems of vibration-related interface compatibility in engine/drive system installations are usually complicated by the inherent coupling of the three major multi-degree-of-freedom systems: engine, airframe, and drive train. The result of this effort is a report documenting dynamic interface problems associated with the CH-54, S-61, CH-53, SH-3, S-58, SH-34, S-64, BLACK HAWK, and the ABC. The ultimate benefit will be the accumulation of data that will eventually lead to a solution of generic problems of this type. This report is one of five reports resulting from engine/airframe/drive train dynamic interface documentation efforts funded by the Applied Technology Laboratory. The related reports and the final report numbers are: Boeing-Vertol, USARTL-TR-78-11; Hughes Helicopters, USARTL-TR-78-12; Sikorsky Aircraft, USARTL-TR-78-13; Kaman Aerospace, USARTL-TR-78-14; and Bell Helicopter, USARTL-TR-78-15.

Mr. Allen C. Royal, Propulsion Technical Area, Technology Applications Division, served as project engineer for this effort.

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The coupled interaction between two or more helicopter subsystems has often been the source of vibration problems - problems often costly and time-consuming to correct because they have not surfaced until the design and development of the individual subsystems is far advanced.

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This report gives a review of Sikorsky experience with such problems over the past twenty years of developing gas turbine powered helicopters. It represents part of an overall Government effort to accumulate data which will eventually lead to solutions of generic problems of this type. The problems presented include forced vibration problems (wherein the excitations come from either aerodynamic loads on the main rotor, or mechanical imbalances in the engine/drive train), self-excited vibrations, and a transient response problem.

The problems are grouped according to type and are presented in chronological order. They include where possible: a brief description of the problem, the investigation leading to a solution, the resulting solution, any limitations associated with the solution, and any unsolved problem areas.

Recent trends in problems are discussed. Recommendations are made for future analytical/ testing efforts to achieve an improved understanding of interfacing dynamic problems and potential solutions.

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PREFACE

The work reported herein was performed by the Sikorsky Aircraft Division of United Technologies Corporation under Contract DAAJ02-77-C-0037 for the Applied Technology Laboratory, U.S. Army Research and Technology Laboratories (AVRADCOM), Fort Eustis, Virginia 23604. The work was carried out under the technical cognizance of Mr. Allen Royal of the Applied Technology Laboratory, AVRADCOM. The program was conducted under the management of Sikorsky Aeromechanics Branch Manager Mr. Peter Arcidiacono and Aeromechanics Chief of Dynamics Mr. William Kuczynski. Sikorsky engineering personnel directly involved in the program include Mr. William Twomey, Mr. Edward Ham, Mr. David Adams, Mr. Richard Barnard, Dr. Raymond Carlson, Mr. Gerard Gardner, Mr. Daniel Hansen, Mr. Robert Kuzma, Mr. Joseph Magri, Mr. Richard McBride, Mr. George Molnar, and Mr. Paul vonHardenberg. Additional help was provided by Mr. R.I. McCormick of Pratt and Whitney Aircraft of Canada, Ltd.

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INTRODUCTION

The coupled interaction between two or more helicopter dynamic subsystems has often been the source of vibration problems. The total helicopter dynamic system has potential sources of excitation in the rotor, fuel control, drive train or airframe systems. Vibration problems related to engine/airframe/drive train interface compatibility are usually complicated by the inherent coupling of the four major multi-degree-of-freedom systems: engine, airframe, drive train, and rotor.

Often these dynamic interface problems are among the last to be seen (and thus the most costly to correct) in the design of a subsystem such as an engine, because they involve the presence of another subsystem or subsystems such as a drive train or an airframe. Thus, the designs of both subsystems are often far advanced before the problem is discovered. In addition, the prediction of these problems is further complicated by the fact that it often requires extremely close interaction between two or more manufacturers and several technical disciplines.

It is desirable to provide a complete report of past gas turbine powered helicopter engine/airframe/drive train dynamic interface problems as part of an overall Government effort to accumulate data which will eventually lead to a solution of generic problems of this type.

With this end in view a systematic review was conducted at Sikorsky Aircraft to uncover and document all such problems encountered within the company over the past twenty years. The review was initiated by requesting all the relevant engineering departments within the company to make up a list of all known problems. This was followed up by interviews with the cognizant engineers and file searches for relevant documentation. Problems not relevant to the present investigation were sifted out. The criterion for a relevant problem was that it involve an interaction between at least two of the subsystems: engine, airframe, and drive train. The engineer who emerged as the most knowledgeable about a particular problem was asked to write it up. These writeups formed the basis of the present report.

Eighteen problems are reported on herein. The contractor's S-76 helicopter was not included in the review.

A summary and classification of the problems are given, followed by detailed descriptions of the problems, and finally observations, conclusions, and recommendations for future work.

SUMMARY OF PROBLEMS

Table 1 gives a summary of the turbine-powered aircraft developed at Sikorsky Aircraft over the past twenty years. In developing and testing these aircraft, various dynamics problems were encountered which involved an interaction between two or more of the aircraft subsystems: engine, airframe, and drive train. Figure 1 gives a summary view of these problems, shown versus the year in which they occurred and the aircraft with which they were associated. The horizontal bar representing each aircraft indicates a five-year period centered on the date of first flight of that aircraft. Most of the problems occurred during this five or six year aircraft developmental period. A total of 18 problems are documented.

Problems 4 and 9, which are seen in Figure 1 to have occurred long after their respective aircraft first flights, were brought on by later modifications made to their engine installations. Problem 18 didn't make its appearance until the more powerful JFTD12A-4A engine was installed in the CH-54. A summary of the problems and their solutions are presented in Table 2.

The 18 problems have been classified according to their underlying generic type:

1. Forced vibrations
   a. Main rotor aerodynamic excitation
   b. Mechanical imbalance excitation
   c. Other excitation
2. Self-excited vibrations
   a. Energy source: fuel control system
   b. Energy source: other
3. Transient response

FORCED VIBRATIONS (10 PROBLEMS)

The vibratory motion of forced vibration problems depends on an external excitation force which is "independent" of the motion. Ten of the 18 problems fell into this category (Problems 1 to 10). The external forces doing the exciting came from either aerodynamic forces acting on the main rotor, mechanical imbalance in the drive train, or in one case, from the rotating teeth of a starter jaw.

In all of these cases the exciting forces caused excessive vibration response in the engine or drive train because of the close proximity of the driving frequency to a natural frequency of the driven component. This natural frequency was usually that of an engine installation, a drive train component, or in one case, that of the rotor/drive train torsional system.

Their solutions, for the most part, involved structural or weight modifications which either raised or lowered the offending natural frequency to move it away from the forcing frequency. In one case the mechanical imbalance exciting force was reduced (Problem 9).

These problems are shown in Figure 2.

Main Rotor Aerodynamic Excitation (6 Problems)

In six of the ten forced vibration cases, the external exciting forces originated in the aerodynamic forces acting on the main rotor blades, as they passed through a nonaxisymmetric flow field (Problems 1 to 6). This nonaxisymmetry in the flow field arises from a superimposed lateral velocity (in forward flight) or from nonaxisymmetric flow boundary conditions due to the presence of the airframe (in hovering flight and ground testing). These blade forces are transmitted through the hub to the remainder of the aircraft at the equal-blade-track frequency of nP (number of blades times the main rotor speed). The resulting forced vibration thus occurs at a frequency of nP. In one case the rotor wake impinged directly on the airframe to cause the excitation, also at a frequency of nP (Problem 3). A summary view of these problems is given in Figure 3.

Mechanical Imbalance Excitation (3 Problems)

In three of the forced vibration problems the motion was excited by mechanical imbalance in either an engine turbine or in a drive train shaft (Problems 7 to 9). The frequency of the motion was the rotational frequency of the unbalanced component. Figure 4 gives a summary view of these problems, which are often referred to as critical speed problems.

Other Excitation (1 Problem)

One forced response problem derived its excitation force from clashing gear teeth in a starter jaw (Problem 10, Figure 5).

SELF-EXCITED VIBRATIONS (7 PROBLEMS)

In self-excited vibrations, the presence of an independent exciting force is not necessary for the motion to continue. It is only necessary to have the presence of an energy source into which the motion can tap and thereby create an alternating force which sustains the motion. The frequency of the self-excited vibration is a natural frequency of the system.

Seven of the 18 problems were classified under this heading (Problems 11 to 17). In these, the self-excited natural mode was most often a mode of the rotor/drive train torsional system, although in one case it was a drive shaft lateral mode (Problem 13).

The time of occurrence of these problems is given in Figure 6. In all but two, the energy source was the fuel control system (fuel flow). In problem 13 the energy source was the internal hysteretic damping in the drive train. In Problem 16, it is not certain whether the energy source came from the fuel control system or from aeroelastic flap-lag-pitch coupling in the main rotor blades.

TRANSIENT RESPONSE (1 PROBLEM)

One dynamics problem (Problem 18) involved transient response rather than vibratory motion (Figure 7).

OBSERVATIONS

INCREASED PREVALENCE OF SELF-EXCITED VIBRATIONS

Among the principal classes of dynamic interface problems, those which show the most marked increase in recent years are the self-excited vibrations (Figure 6). In the years prior to 1973, only 3 of the 12 problems were of this type (Problems 11 to 13). Since then, however, 4 out of 6 problems have been of this type (Problems 14 to 17). The highly coupled nature of self-excited oscillations usually makes them more difficult to analyze. The excited component cannot be treated separately from the exciting force as it often is in forced response problems.

In contrast to these, the rate of occurrence of forced vibration problems has not varied greatly over the years (see Figure 2).

The energy source in most of the self-excited problems was in the fuel control system, indicating, probably, the higher performance (higher gain) being asked of recent engine/aircraft systems. For example, the solution in Problem 17 was partly a compromise between sufficient "torsional" stability margin on the one hand and sufficient responsiveness of the propulsion system in severe maneuvers, such as jump takeoffs and recovery from autorotation, on the other hand.

INCREASED PREVALENCE OF PROBLEMS INVOLVING THE ROTOR/DRIVE TRAIN TORSIONAL SYSTEM

Another trend stands out if one looks over the type of motion, or mode shape, involved in the various problems of Table 2. Prior to 1972, most of the problems involved fairly localized motions. They involved either engine-on-airframe modes, wherein the airframe could usually be considered rigid (Problems 1 to 5), or local high frequency component modes (Problems 7 to 10, and 13). Solutions were obtained for most of these problems using simple localized analyses and tests. Main rotor blade degrees of freedom, for example, were not needed in the models used to analyze them.

Of the six problems which have occurred since 1972, however, at least three involved coupled modes of the complete rotor/drive train torsional system (Problems 6, 16, and 17). In two of these three problems, main rotor blade flexibility (in addition to rigid blade motion) was an essential ingredient in the critical torsional mode (Problems 6 and 16). Prior to 1972, only one problem involved this torsional system (Problem 12). This is shown in Figure 8.

RECENT IMPORTANCE OF MAIN ROTOR BLADE LAG DAMPING

Damping available from the main rotor blade lag dampers was not important in any of the dynamic interface problems prior to 1972. Since then, however, it has been an important factor in all three of the recent problems involving rotor/drive train torsional modes (Problems 6, 16, 17). See Figure 9. The variation in lag damping available with aircraft forward speed (and to a lesser extent with other flight conditions) tended to be the critical parameter.

INCREASED PREVALENCE OF PROBLEMS INVOLVING FUEL CONTROL SYSTEM

Three of the six problems since 1972 required inclusion of the fuel control system equations to explain the phenomena and arrive at a solution (Problems 14, 16, and 17). Prior to 1972 only two vibration problems did (Problems 12 and 18). See Figure 10.

TABLE 1. SIKORSKY TURBINE-POWERED HELICOPTERS

Helicopter	Design Gross Weight(lb)	No. of Main Rotor Blades	No. of Engines	Engine Manufacturer	Engine	No. of Gearboxes in Drive Train
SH-34H	10,500	4	2	General Electric	T58	4
HH-52	7,500	3	1	General Electric	T58-GE-8	3
SH-3A (S-61)	16,200	5	2	General Electric	T58-GE-10	3
S-61 (COMM)	20,500	5	2	General Electric	CT58-140-2	3
CH-54	38,000	6	2	Pratt & Whitney	JFTD12-5A	5
CH-53	33,500	6	2	General Electric	T64-GE-6	5
S-67	17,300	5	2	General Electric	T58-GE-5	3
S-58T	13,000	4	2	Pratt & Whitney of Canada	PT6T-6	4
XH-59A	9,800	3 & 3 (coaxial)	2	Pratt & Whitney of Canada	PT6T-6	1
CH-53E	45,000	7	3	General Electric	T64-GE-415	5
YUH-60A	16,450	4	2	General Electric	T700-GE-700	5
S-72	18,400; 26,200	5	2	General Electric	T58-GE-5	3

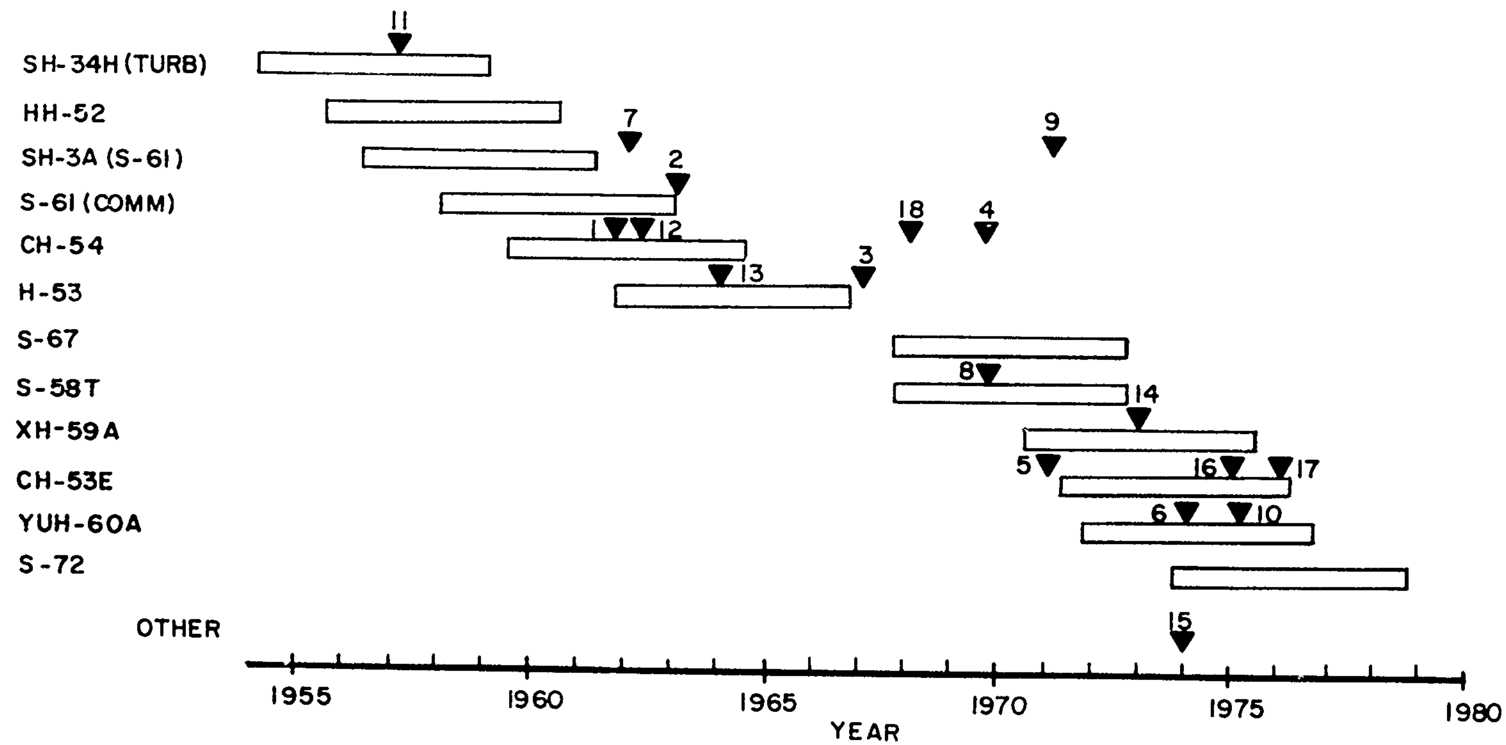


Figure 1. Engine/Airframe/Drive Train Dynamic Interface Problems vs Aircraft and Year.

TABLE 2. SUMMARY OF PROBLEMS

PROBLEM DESCRIPTION	AIRCRAFT	YEAR	SOLUTION
Forced Vibrations, Main Rotor Aero. Excitation, at nP:			
1) Engine rigid body roll mode (ω_n = 5.8P) excited at 6P	CH-54 (PSTB)	1962	Installed soft spring between torquemeter and airframe to lower engine natural frequency to 3.2P
2) Engine 1st bending mode (ω_n = 6.8P) excited at 5P	S-61L	1963	Replaced "hard" front engine mounts with "soft" elastomeric mounts to lower bending frequency to 2.9P
3) Engine roll/yaw/lat. mode excited at 6P in downwind towing	CH-53A	1967	Limited tow operation to 100 hours
4) Engine 1st bending mode (ω_n = 7.6P) excited at 6P	CH-54B	1970	Added a spring between airframe and engine to raise bending frequency to 8.8P
5) Engine longitudinal mode (ω_n = 7.35P) excited at 7P	CH-53E (PSTB)	1971	Stiffened controlling gimbal ring to raise natural frequency to 15.9P
6) Rotor/drive train 3rd torsional mode (ω_n = 4.1P) excited at 4P	YUH-60A	1974	Added edgewise stiffening to main rotor blades to raise natural frequency to 4.4P

TABLE 2. (continued)

PROBLEM DESCRIPTION	AIRCRAFT	YEAR	SOLUTION
Forced Vibrations, Mechanical Imbalance Excitation, at $N_{component}$:			
7) Engine torque tube bending mode (ω_n = 100% N_F) excited at N_F	SH-3A	1962	Attached 12½-lb wt to torque tube to lower natural frequency to 85% N_F
8) Engine drive shaft critical speed problem (ω_n = 127% N_{shaft}), excited at N_{shaft}	S-58T	1970	Replaced steel shaft with aluminum to raise critical speed to 144% N_{shaft}
9) Engine drive shaft/torquemeter critical speed problem, excited at N_{shaft}	NSH-3A	1971	Removed pole piece torquemeter to reduce imbalance
Forced Vibration, Other Excitation:			
10) IPS shaft torsional mode (ω_n = 180 Hz) excited by rotating starter jaw	YUH-60A	1975	Reprogrammed starter jaw speed-buildup so that jaw made contact at lower speed
Self-Excited Vibrations:			
11) Engine load sharing oscillation (< 1 Hz)	S-58T	Late 1950's	Increased N_F governor droop slope from 2% to 6-8%

TABLE 2. (concluded)

PROBLEM DESCRIPTION	AIRCRAFT	YEAR	SOLUTION
12) Feedback torsional oscillation of N_F cable mode (11 Hz)	CH-54	1962	Shortened N_F cable to raise natural frequency
13) Drive shaft supercritical hysteresis oscillation (33 Hz)	CH-53A	1964	Replaced Zurn coupling by frictionless Thomas coupling
14) Engine hunting oscillation	XH-59A (PSTB)	1973	Reduced governor gain and removed torque equalizer
15) Minor engine hunting oscillation in accessory drive operation only (0.5 - 0.7 Hz)	-	1974	Situation accepted by customer as "standard to type" since not detrimental
16) Oscillation of rotor/drive system 3rd torsional mode (3.6P)	CH-53E	1975	Softened main rotor blade edgewise stiffness to lower natural frequency to 3.45P, decoupling it from flapping frequency and increasing modal motion across lag damper
17) Feedback oscillation of rotor/drive system 1st torsional mode (2 Hz)	CH-53E	1976	Increased N_f governor time constant to lower fuel control gain at 2 Hz
Transient Response:			
18) Excessive transient droop	CH-54	1968	Redesigned fuel control system to put engine on max. accel. schedule at smaller speed errors (3% instead of original 10%)

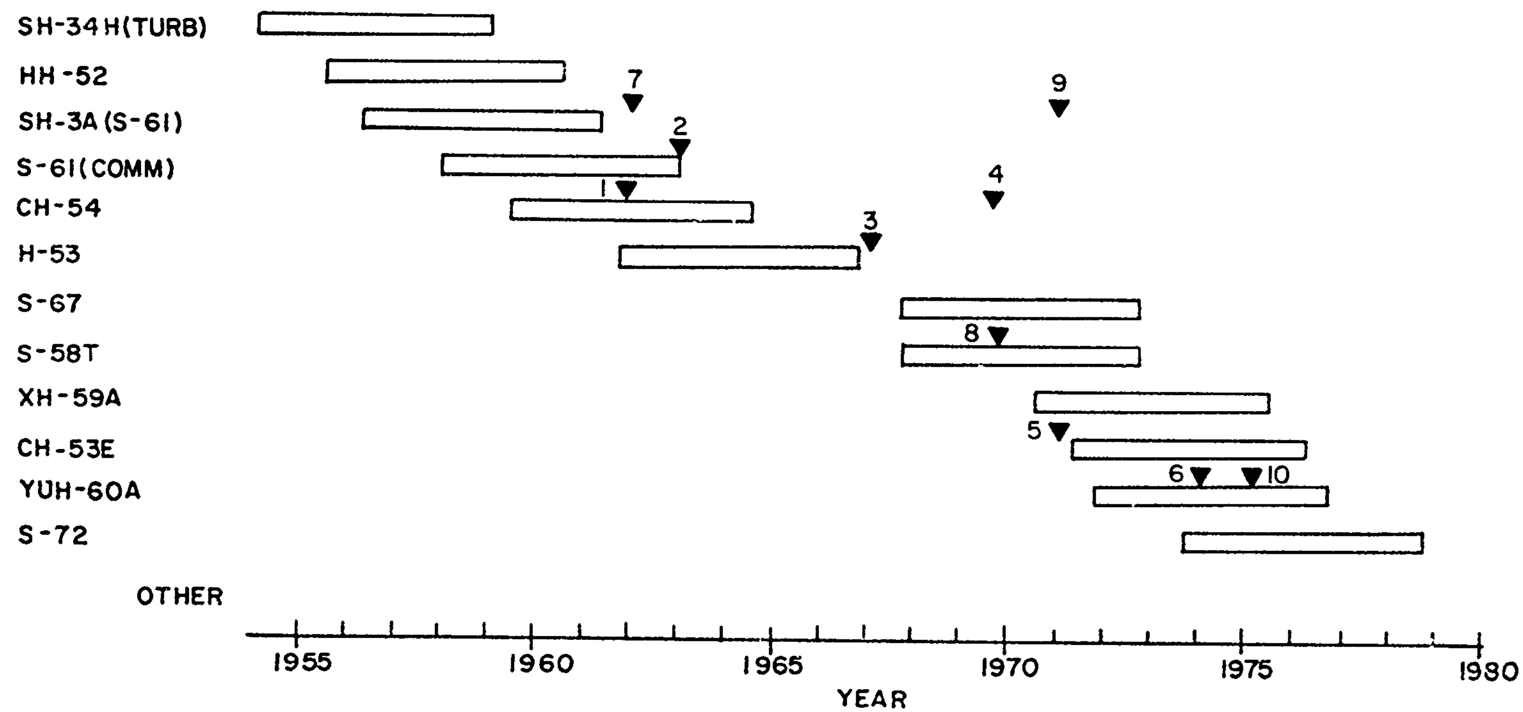


Figure 2. Forced Vibration Problems.

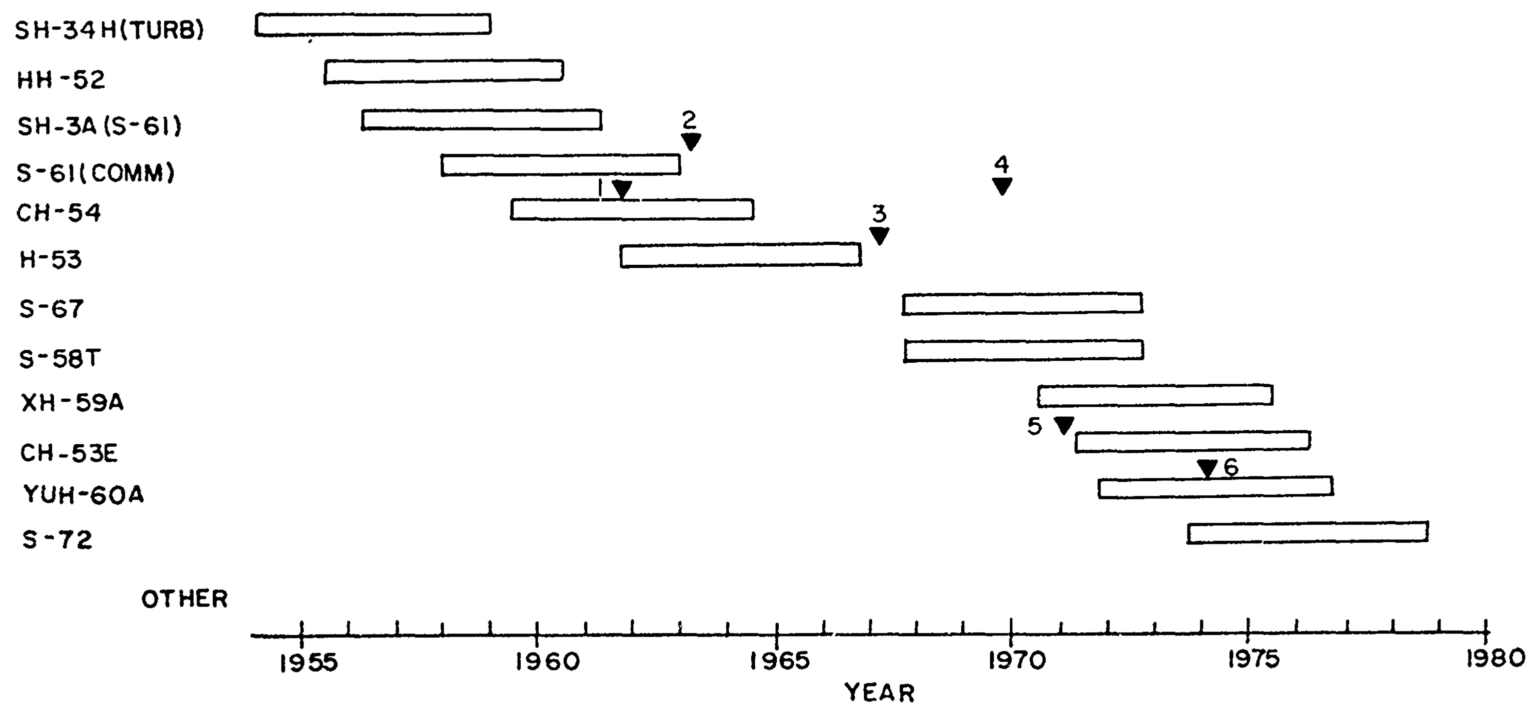


Figure 3. Forced Vibration Problems, Main Rotor Aerodynamic Excitation.

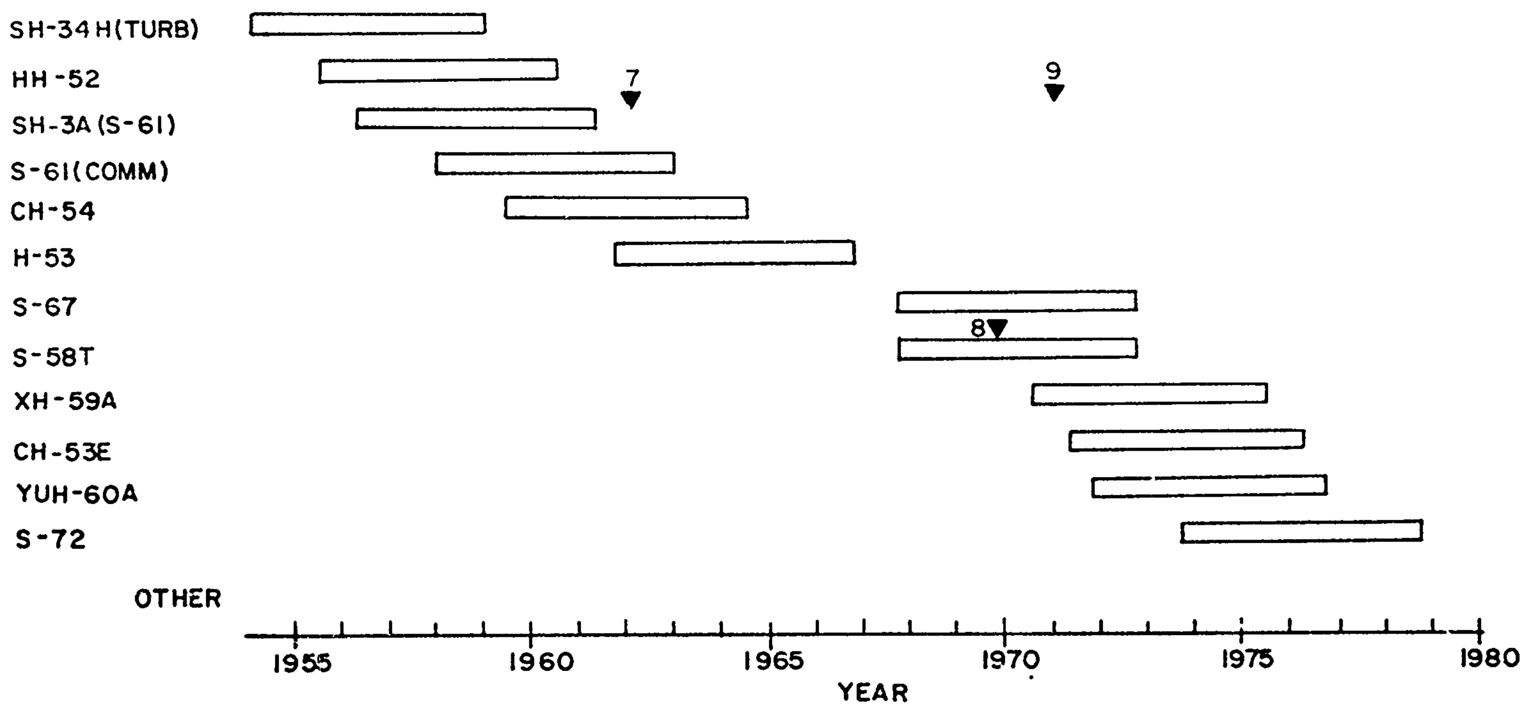


Figure 4. Forced Vibration Problems, Mechanical Imbalance Excitation.

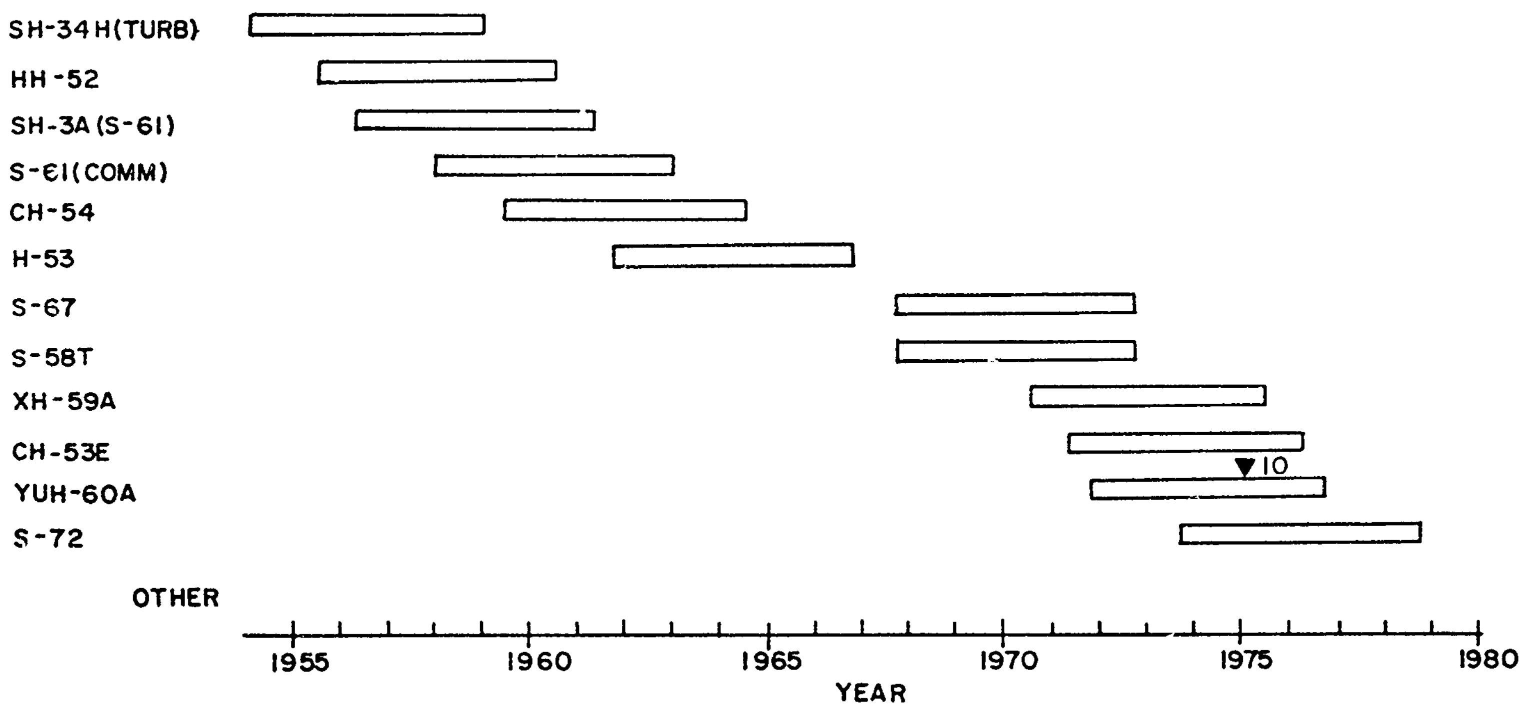


Figure 5. Forced Vibration Problems, Other Excitation.

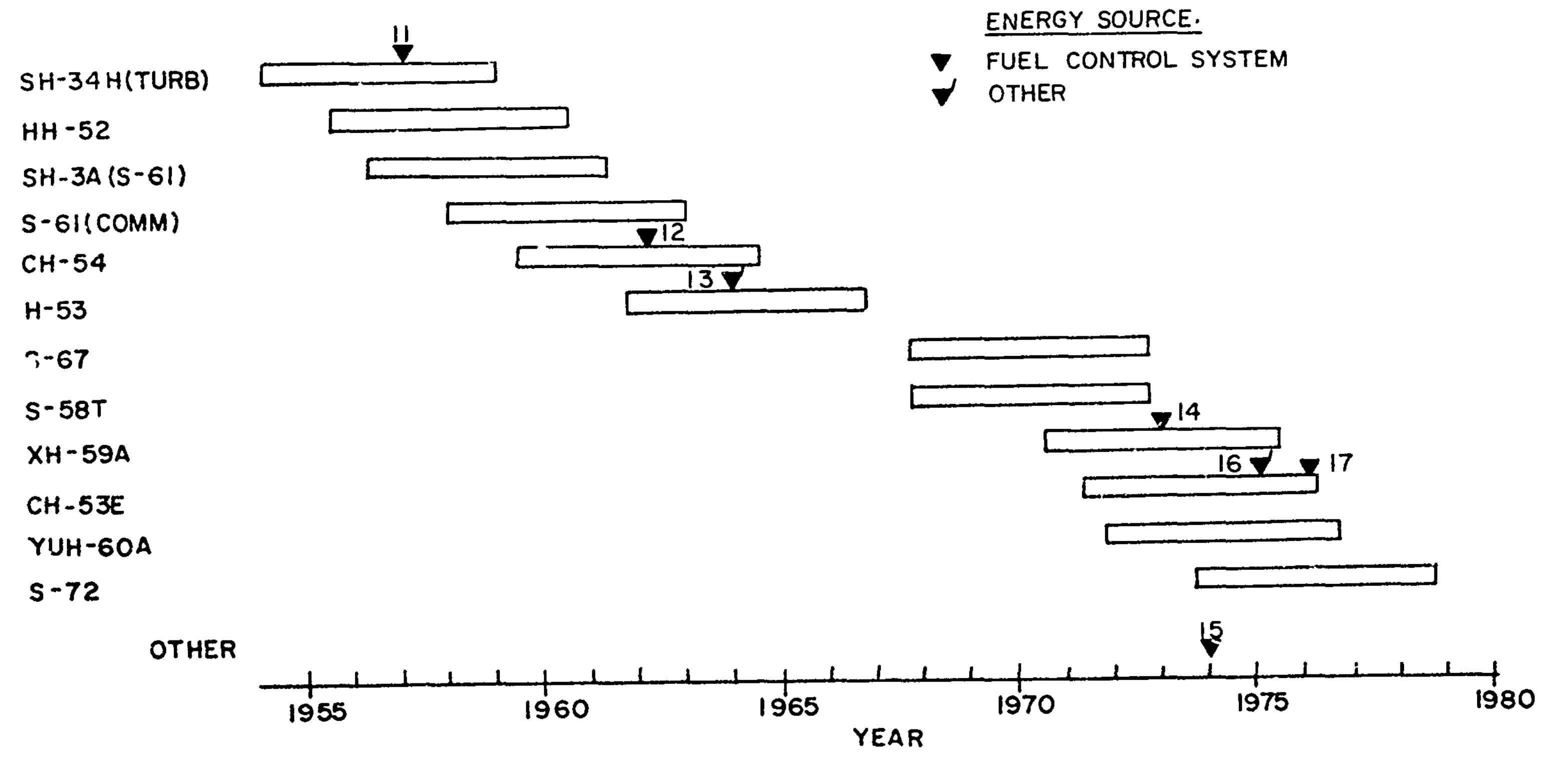


Figure 6. Self-Excited Vibration Problems.

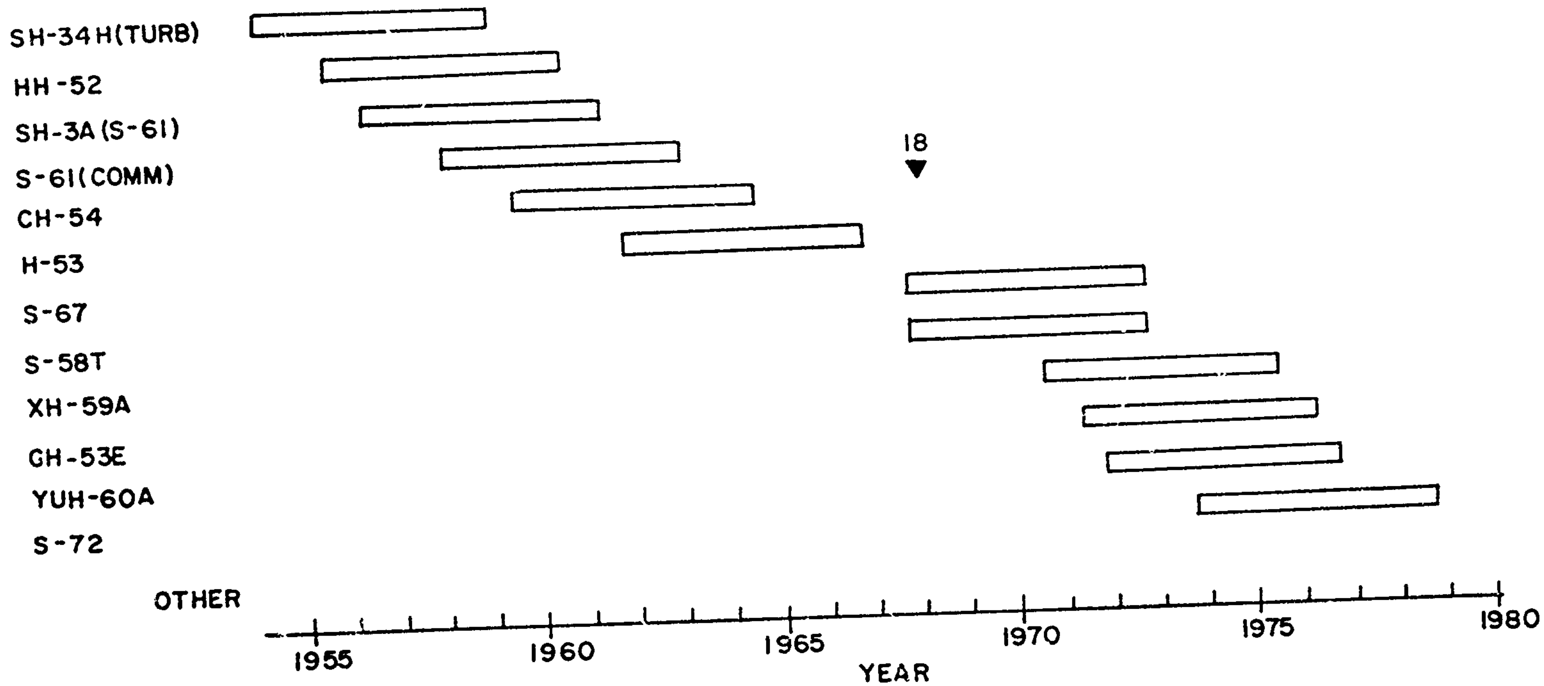


Figure 7. Transient Response Problems.

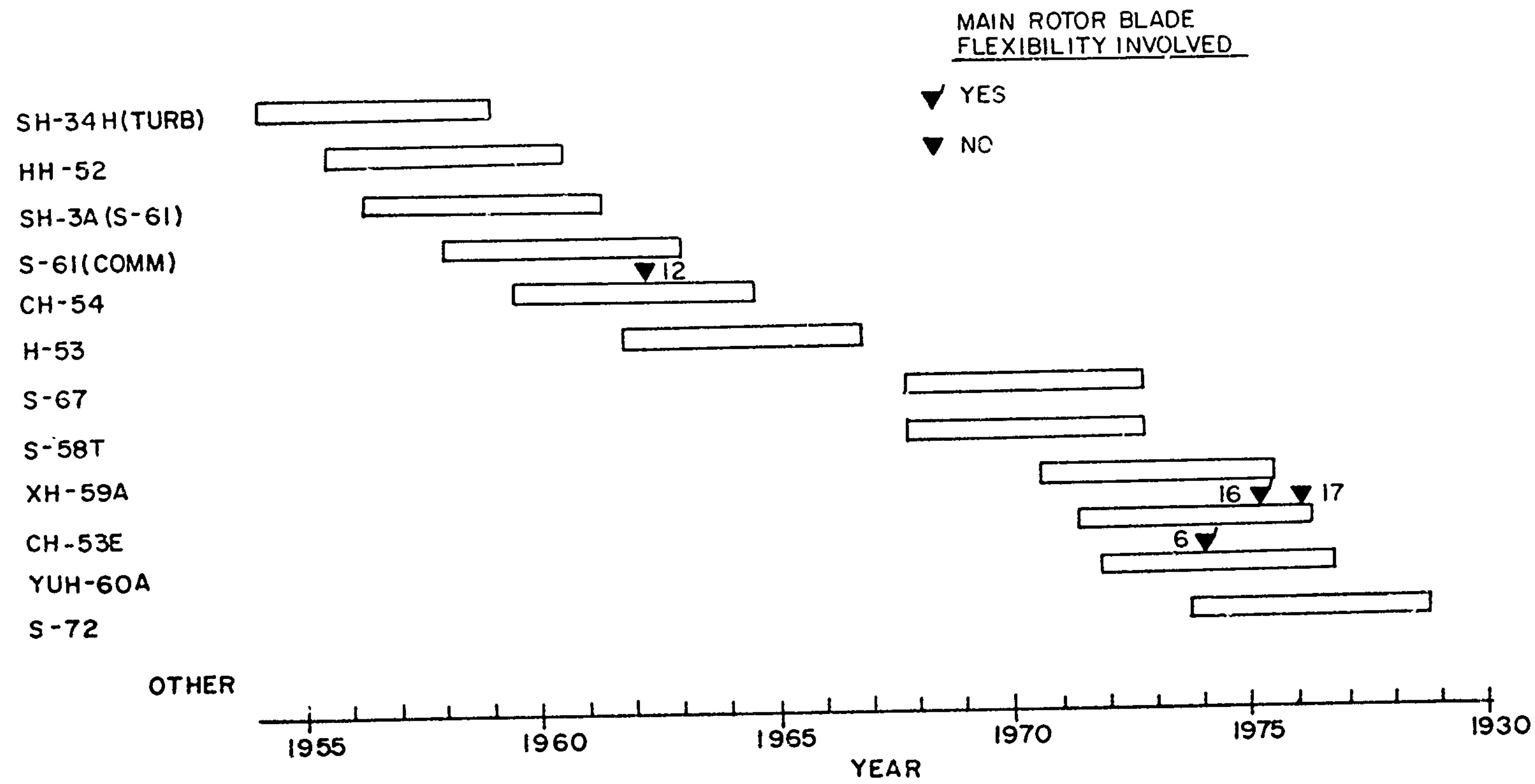


Figure 8. Problems Involving Rotor/Drive Train Torsional System.

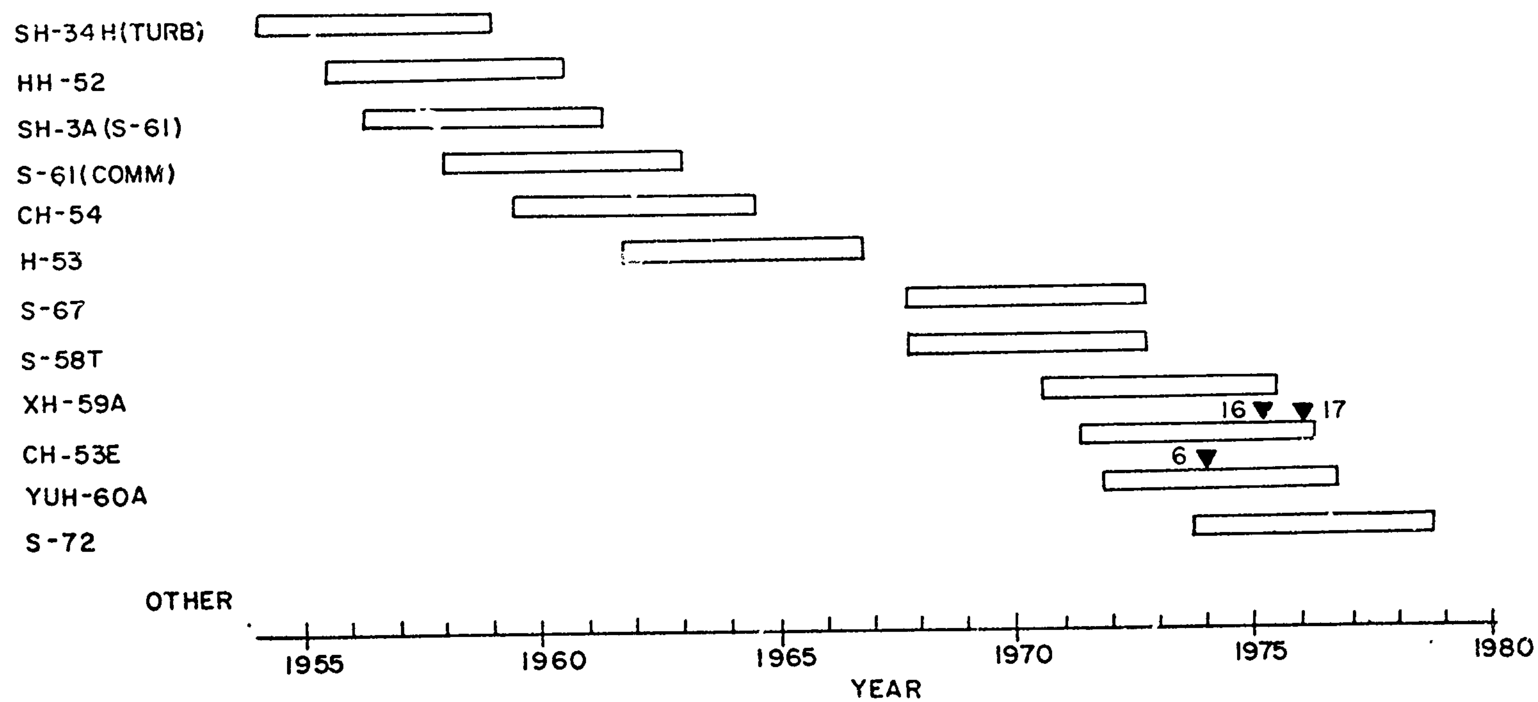


Figure 9. Problems Involving Main Rotor Blade Lag Damping.

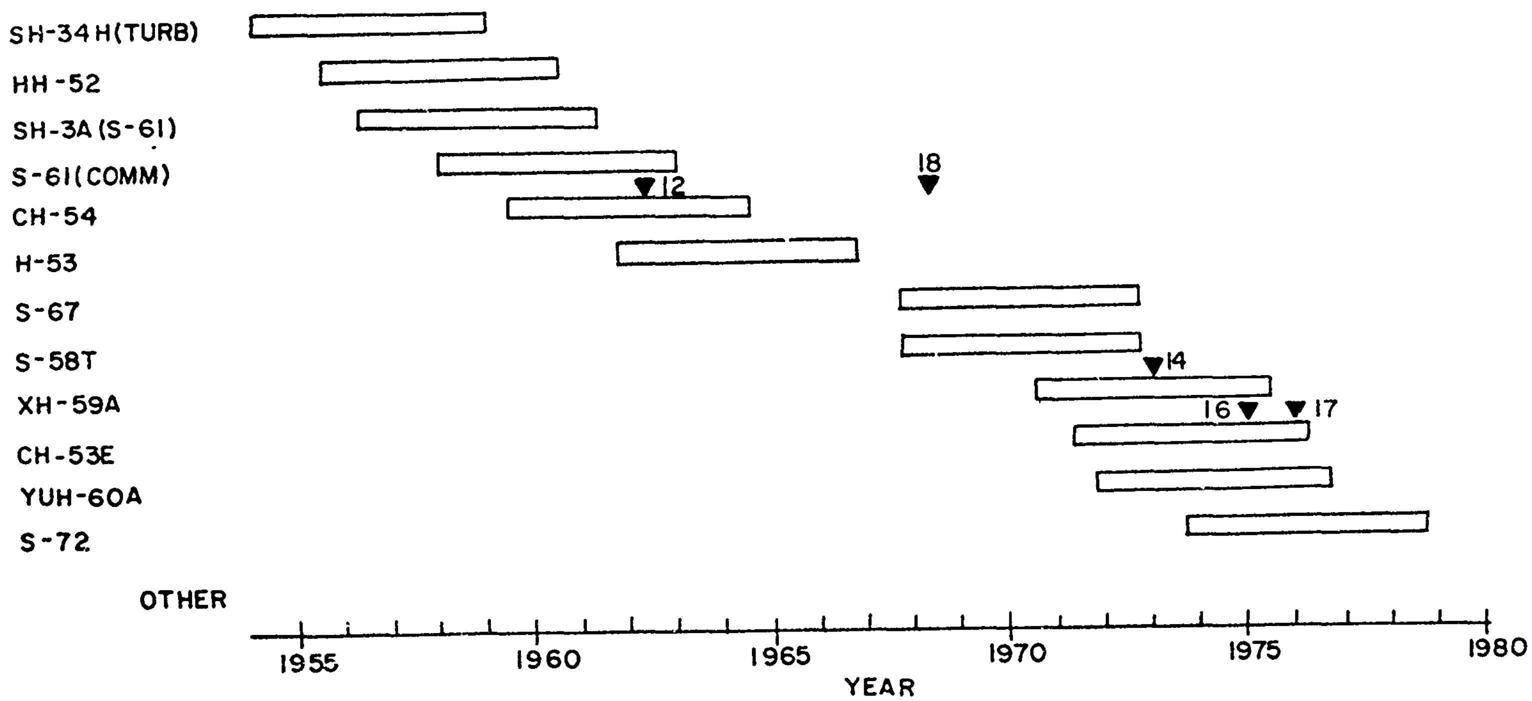


Figure 10. Problems Involving Fuel Control System.

DETAILED PROBLEM DESCRIPTIONS

FORCED VIBRATION PROBLEMS, MAIN ROTOR AERO. EXCITATION

Problem 1. CH-54/JFTD12: Engine Roll Mode Excited at 6P

Problem Manifestation

The problem was first detected while conducting the initial Ground Test Program on the Propulsion System Test Bed (PSTB) for the model S-64 (later designated CH-54A for the Army). The engine was a Pratt & Whitney JFTD12 free turbine engine. The problem was identified in early 1962.

First indication of the problem was the observance of large (± 1200 SHP) fluctuations in the engine torquemeter. It was determined that these fluctuations were not actual engine power fluctuations but were caused by torsional vibration of the engine case which was being sensed by the reaction-type torquemeter system in use at that time. Figure 11 is a schematic of the engine installation and torque measuring system.

Had this problem not been resolved, the consequences would have been:

1. Loss of accuracy of the engine torquemeter system.

2. Low reliability of the torquemeter system.

3. Low reliability of engine-mounted equipment such as cooler, fuel control, fuel and oil lines and brackets, etc.

4. Possible cracking of the compressor and/or free turbine case.

Nature of Problem

Investigation revealed the nature of the problem to be a forced response/natural frequency problem where the excitation force was the 6 x main rotor (18.5 Hz) vibration and the rigid body torsional natural frequency of the engine installation was approximately 18 cps. The amplitude of the vibration was a function of the N-per-rev rotor vibrations which on the test bed were highest at high main rotor power. This problem was never evaluated on a flight test helicopter because a solution was developed before the first S-64 aircraft was manufactured.

Solution

Since the N-per-rev vibration amplitude of the main gearbox was not considered to be excessive, the obvious solution for this problem was to shift the rigid body torsional natural frequency of the engine so it would not coincide with the 1 x main rotor (3 Hz) or 6 x main rotor (18.5 Hz) forcing frequency. The engine installation torsional mode was shifted to slightly less than 10 Hz by installing an 11,000 lb/inch spring between the torquemeter load cell and the airframe (see Figure 1). A simple compression spring consisting of laminations of rubber and metal shims was installed.

The desired spring rate was developed by test; that is, the number of laminations was varied until the desired spring rate was obtained.

Subsequent tests conducted with the spring installed demonstrated that the engine case torsional vibration was reduced by a factor of more than 6:1.

It should be pointed out that the torsional vibration was an engine case mode and was not observed in the power drive train (engine drive shaft or transmission shafting) to any measurable degree.

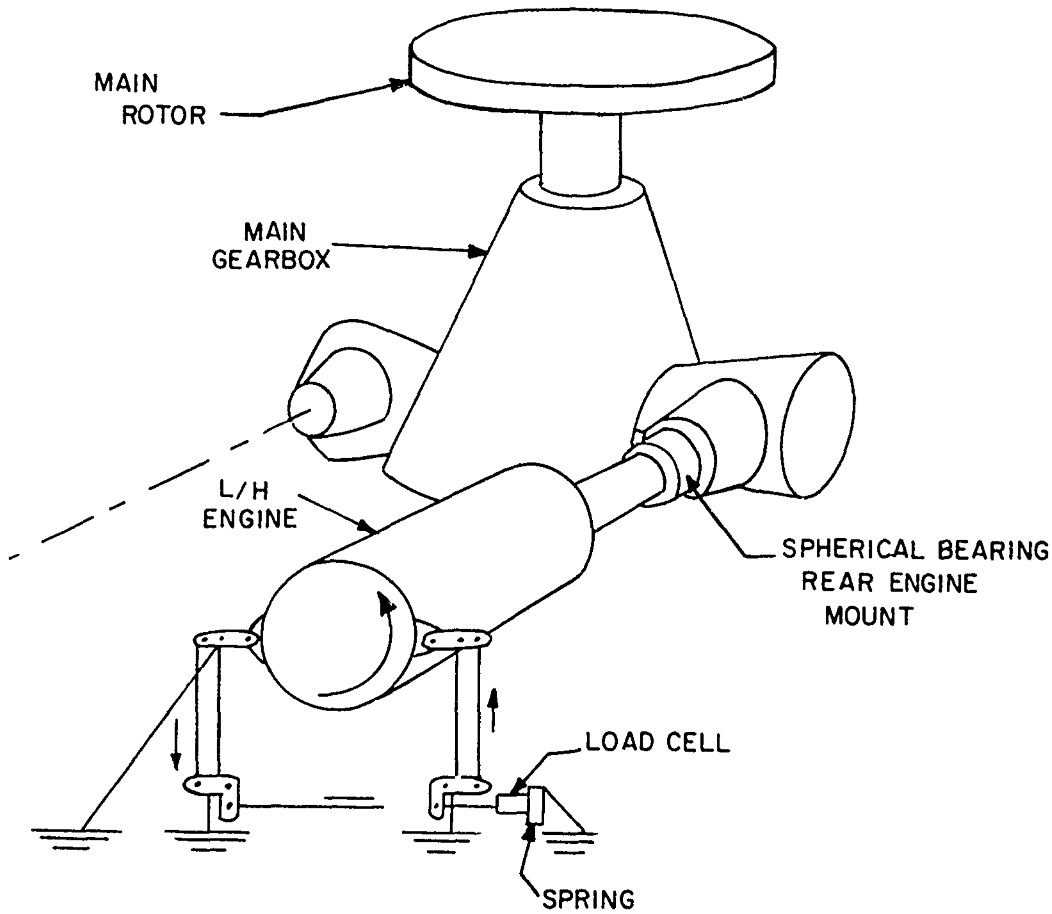


Figure 11. Schematic of Reaction-Type Torquemeter System.

Problem 2. S-61L/T58: Engine 1st Bending Mode Excited at 5P.

Manifestation of Problem

The first indication of this problem in the S-61L was the measurement of engine vibration levels which were above the maximum allowable, according to the engine environment specifications. The problem occurred in 1963 during developmental testing of the S-61L, which was a commercial derivative of the SH-3A. It was known from earlier studies of the SH-3A that these vibratory levels were high enough to cause a high stress problem in the power turbine bearing housing structure.

Nature of Problem

The principal change from the SH-3A, which did not have this problem, was a lengthening of the fuselage by 3 ft to adapt it to its commercial role. The engine mounting system remained unchanged from the SH-3A, with the front frame of the engine mounted directly to the airframe.

Subsequent investigation determined that the increased fuselage length had changed the fuselage response shape, at the main rotor excitation frequency of 5P (17 Hz). In the SH-3A, the problem did not occur because the front frame of the engine was mounted near a node in the airframe response shape. In the S-61L, this node moved away, causing additional transmission of main rotor excitation to the engines. The engine primarily responded in its first bending mode, whose natural frequency was 23 Hz.

Solution

The solution was to increase the interface flexibility between the front frame of the engine and the airframe. Figure 12 shows the S-61L engine installation. The "hard" front engine support struts were switched to "soft" elastomeric isolation mounts. This lowered the resonant frequency of the critical first bending mode from 23 Hz to 10 Hz (Figure 13).

The solution also lowered the 2nd engine bending mode frequency so that now it became the primary contributor to the engine response (Figure 13). The result was an engine response still exceeding the original specifications (which were made in terms of local translational accelerations), but which now caused no excess engine bending (i.e., stress) levels, as the original excessive response had done. As a result of this finding, the engine vibratory response specifications were also changed. The maximum allowable translations were raised, but specifications were added limiting the engine flexural bending angle response to typical values known to have been reached on the problem-free SH-3A. Engine bending, not engine translation, was directly related to the critical stress levels in the power turbine bearing housing structure.

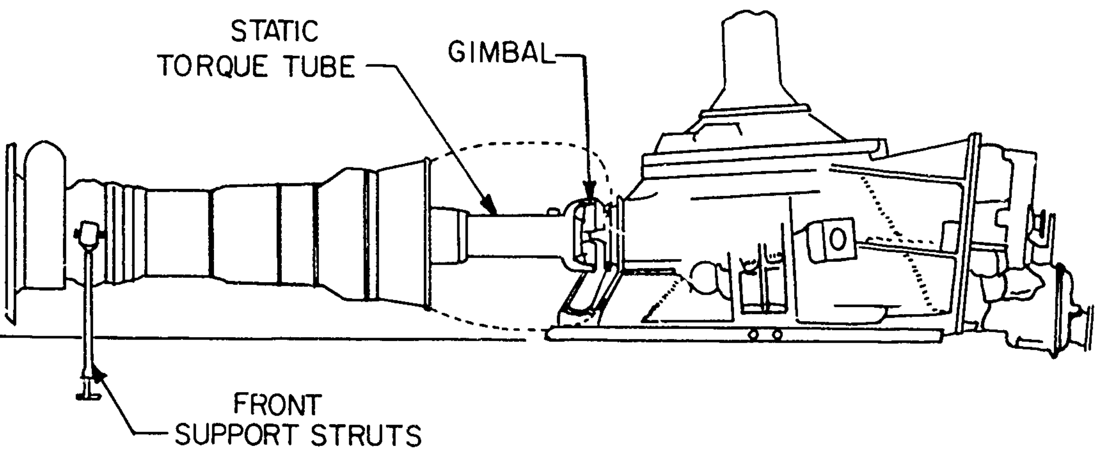


Figure 12. S-61 Engine Installation.

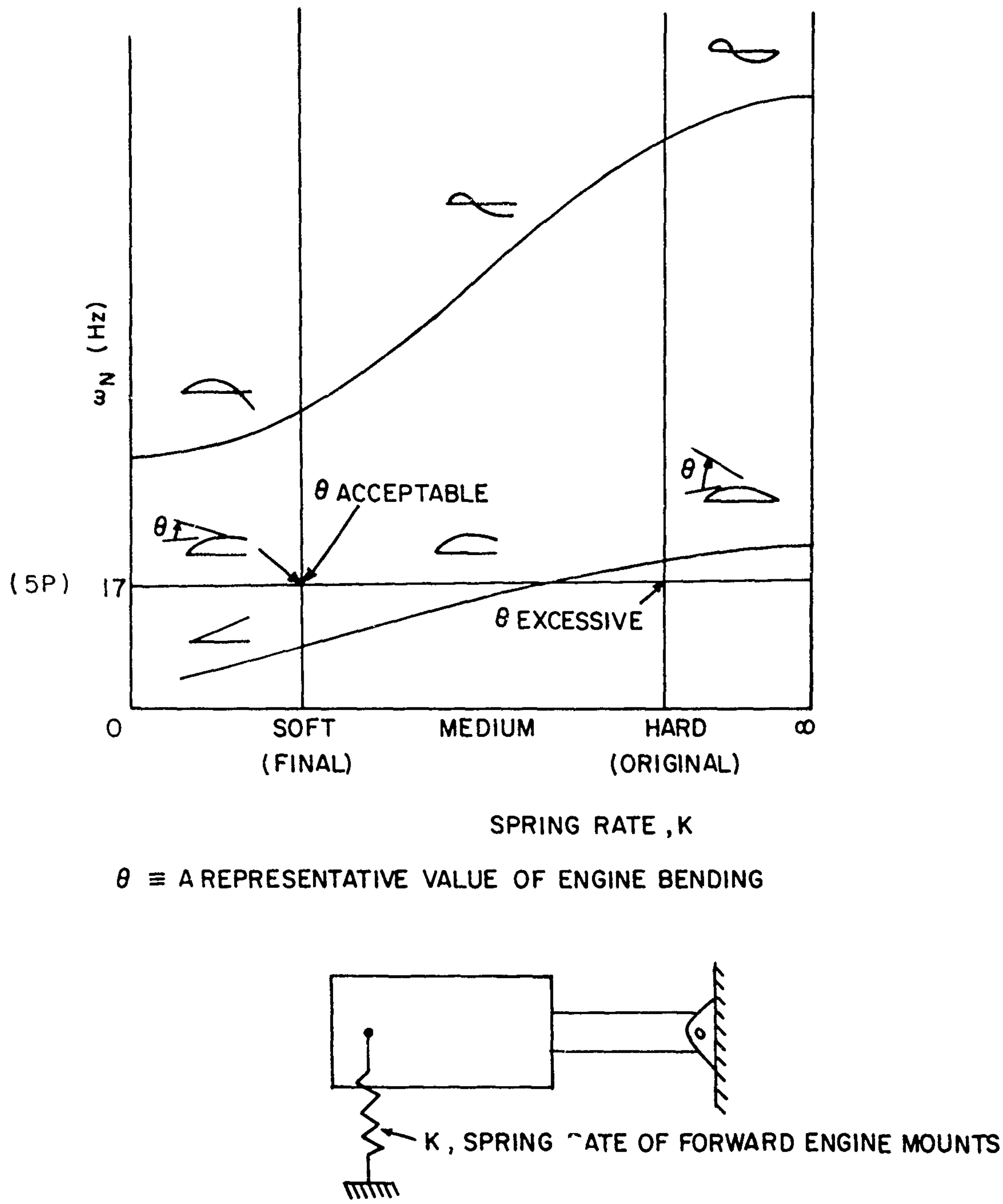


Figure 13. Engine Mode Frequencies vs Spring Rate of Forward Mounts.

Problem 3. CH-53A/T64: Engine Roll Mode Excited at 6P in Downwind Towing

Problem Manifestation

The CH-53A aircraft was initially designed as a transport aircraft. Later, the mine countermeasures mission (MCM) was also assigned. During testing of this mission in 1967, it was discovered that engine vibration levels increased with large tow loads. Under the worst conditions of large tow loads and aft wind direction, vibrations were measured at levels above the engine specification limits and were attributed to forced response caused by the main rotor 6P (18.5 Hz).

Nature of Problem

Under conditions of MCM operations, large tow loads require large main rotor torque. The forward speed is low, and this produces a condition in which large downwash impinges on the fuselage and causes large vibration response in the airframe and engines. Under normal flight operations, large values of main rotor torque occur in high forward speed. At high speed, the downwash is moved aft by the forward velocity and results in airframe/engine excitation.

Engine vibrations were measured to be above specification on the forward mount lateral and the aft mount vertical. The engine roll component was also measured and was strongly coupled with yaw/lateral. It was believed that this coupled mode was near resonance and responding to the main rotor 6P excitation.

Solution

In subsequent meetings of Naval Air Service Command, Sikorsky Aircraft, and General Electric, the data were reviewed and it was decided that towing could be done under the following restrictions:

1. Engine vibration limits were raised from 0.525g to 0.75g, at 18.5 Hz, since the engine vibration response shape in the aircraft installation indicated that the control point acceleration levels could be increased without damage to the engine.

2. Towing limited to 50 hours with engine vibration raised to 1.5g at 18.5 Hz.

3. Main rotor kept at 105% N_R.

4. Engines limited to 100 hours of tow operations.

Problem 4. CH-54B/JFTD12A-5A: Engine 1st Bending Mode Excited at 6P

Problem Manifestation

The CH-54B aircraft had engine forward and aft mounts which were modified from those of the earlier CH-54A model. Analysis of CH-54B flight test data revealed that, during the approach-to-hover flight condition, the number 1 engine vertical vibration exceeded the Pratt and Whitney specification limits. This over-spec condition was discovered during flight testing in January 1970.

Nature of Problem

The vibration problem was that of main rotor excitation of the number 1 engine vertical bending mode. The excitation frequency was 18.5 Hz, which corresponds to 6P, and the engine bending modal frequency was in the 23-24-Hz range as measured on three different aircraft. During flight test operations the measured vibrations were within specification limits except for the approach-to-hover condition. In this condition, the vertical vibration amplitude on the "G" flange was measured to be in the 100-113-mil range, at 18.5 Hz. The specification limit at 18.5 Hz is 50 mils.

Problem Investigation

Modal analysis indicated that the addition of a relatively flexible restraint, mounted between the airframe and free turbine flange, would significantly increase the first vertical bending mode natural frequency of the engine and therefore reduce engine response to the 18.5-Hz main rotor excitation.

Flight testing was conducted with vibration pickups on both engines as sketched in Figure 14. Test conditions consisted of the aircraft at 47,000 lb gross weight with forward and aft c.g. locations, engine torque sweeps, flight in hover normal approach to hover, and trim level flights at various speed and RPM settings. Flights were made with engine basic design installations and also with vertical springs installed between the "G" flanges and airframe. Spring rates were varied from approximately 4500 to 10,000 lb/inch, as estimated from engine modal analysis. Testing results indicated that the least amount of engine vibration under approach-to-hover conditions was achieved with a 7700-lb/inch spring installed. Pratt and Whitney engine personnel expressed a preference for the spring to be located at the engine "H" flange. Flight measurements were taken with the 7700-lb/inch spring at the "H" flange, and similar satisfactory results were obtained. Spring installation at the "H" flange on one engine is shown in the photograph (Figure 15). Presented in Figure 16 are engine relative vertical bending mode shapes for the design installation and for the configuration with added 7700-lb/inch spring at the "H" flange.

Ground vibration tests were conducted on the engines of three different aircraft to obtain comparative data between the basic installation and the 7700-lb/inch added spring installation. A unidirectional shaker was used to excite the engines in the vertical direction. Presented in Figure 17 are the three "G" flange vertical frequency response curves from the basic engine installation which show the resonant frequency range from 23 to 24 Hz. A comparison of "G" flange vertical frequency responses between the basic installation and the added spring configuration is shown in Figure 18. The effect of adding the spring was that of increasing the bending mode frequency from 23-24 Hz to 27 Hz.

Solution

The solution was to install spring restraints having a vertical spring rate of 7700 lb/inch at the engine "H" flange. Production aircraft were delivered in this configuration and there were no further problems.

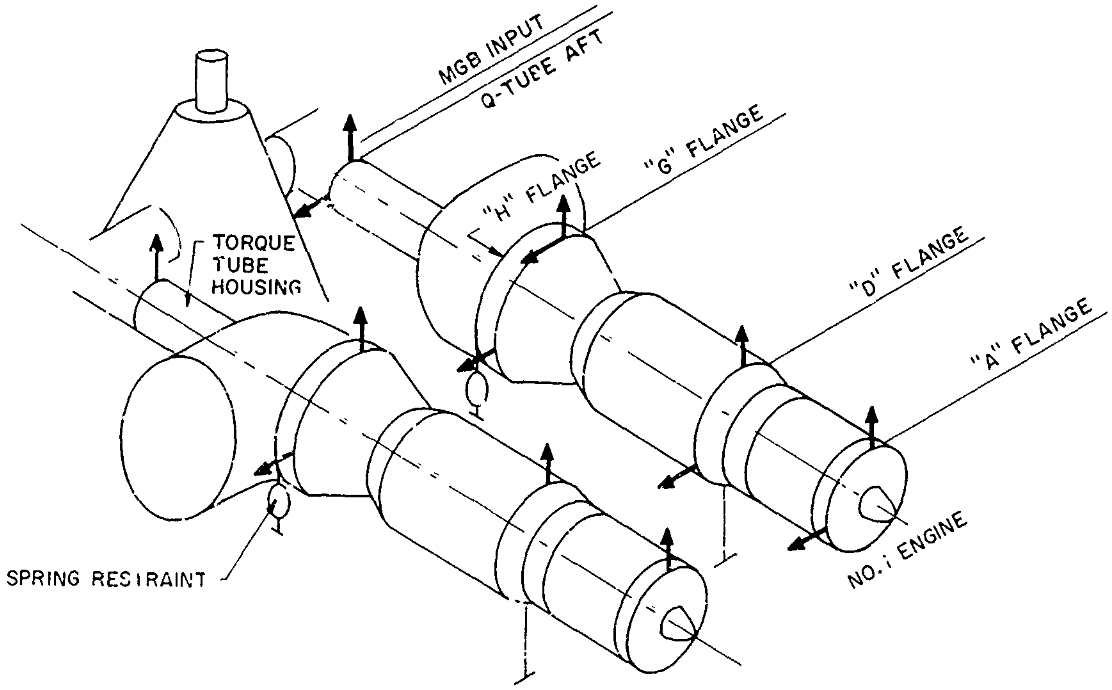


Figure 14. Engine Vibration Pickup Locations.

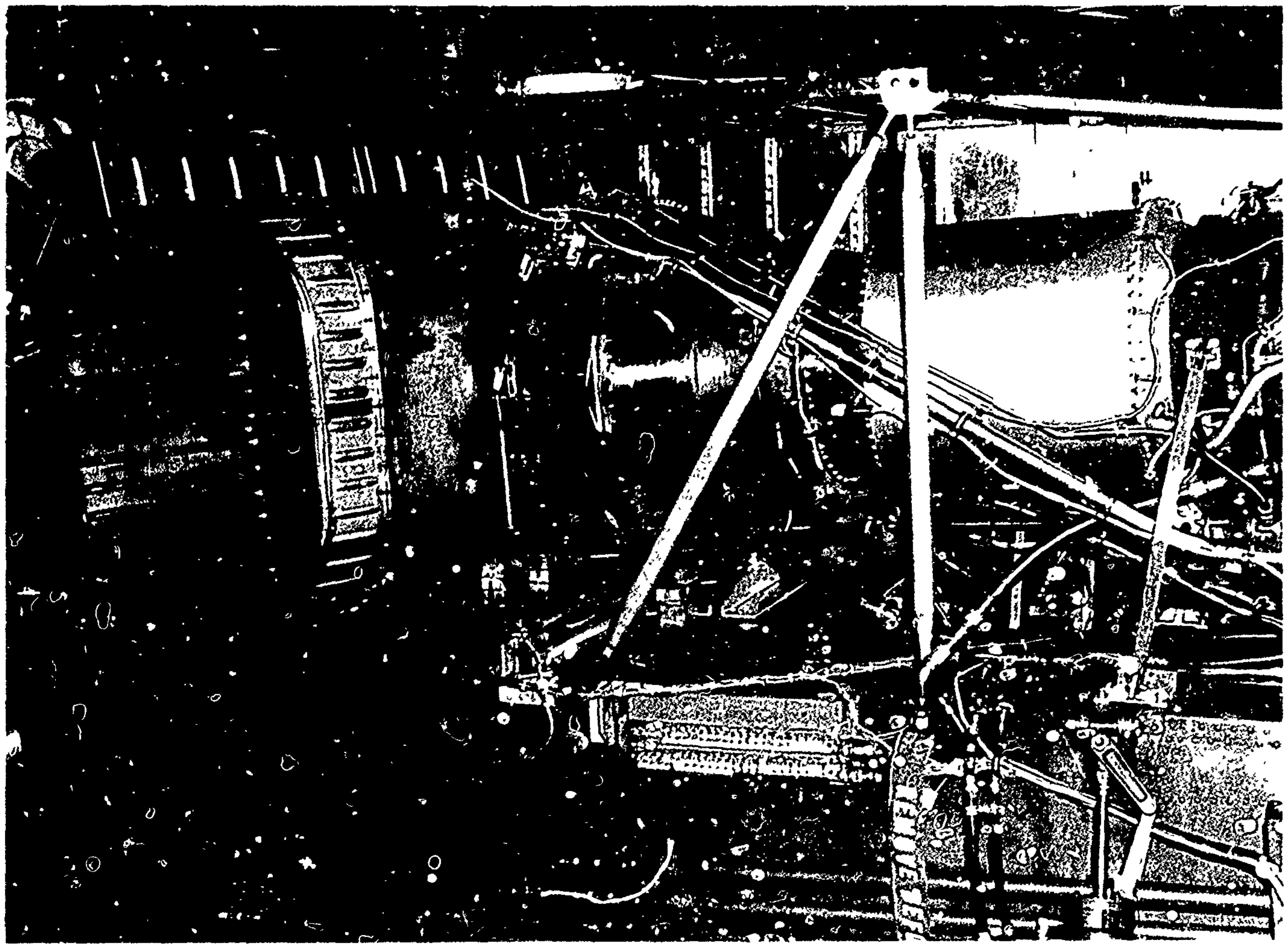


Figure 15. Spring Installation at the Engine "H" Flange.

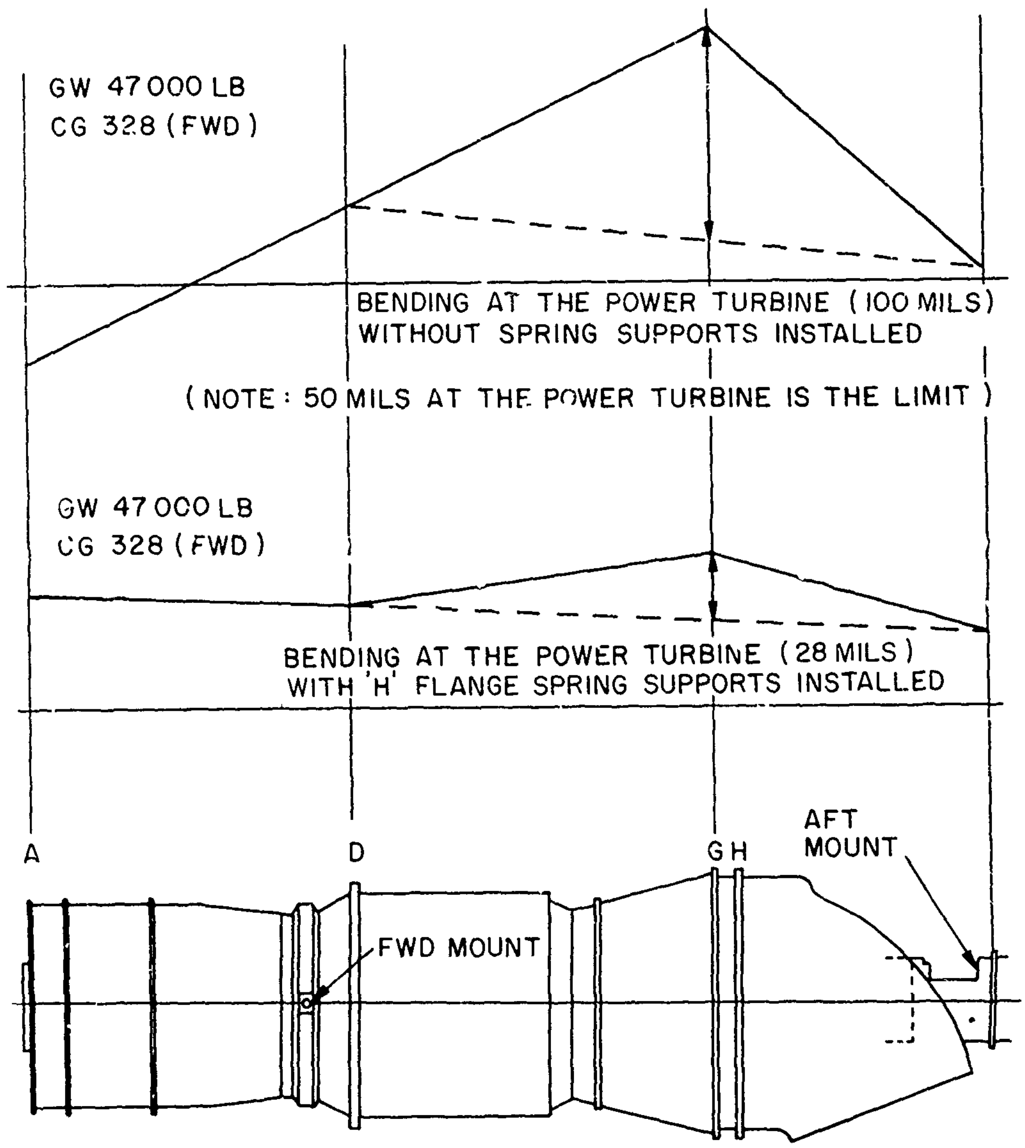


Figure 16. Engine Vertical Mode Shapes.

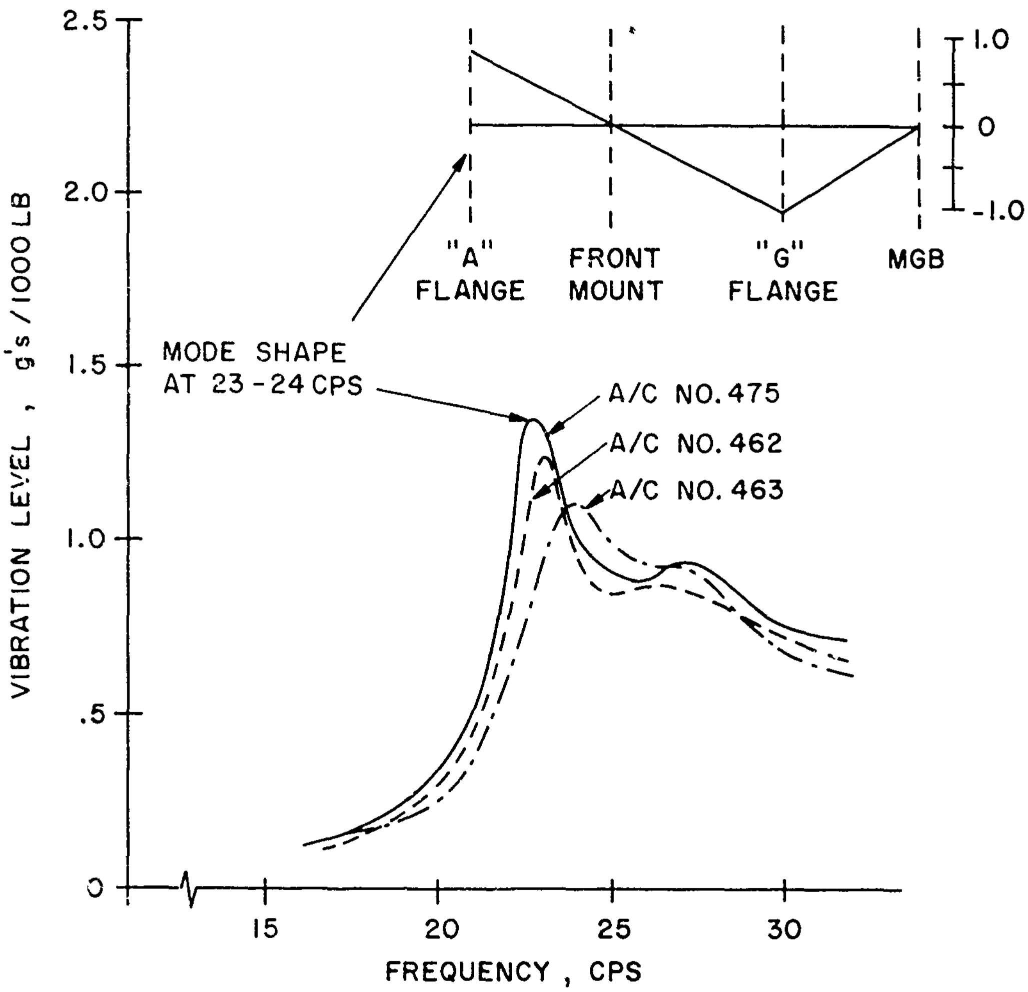


Figure 17. Power Turbine Vertical Response vs Shaker Frequency.

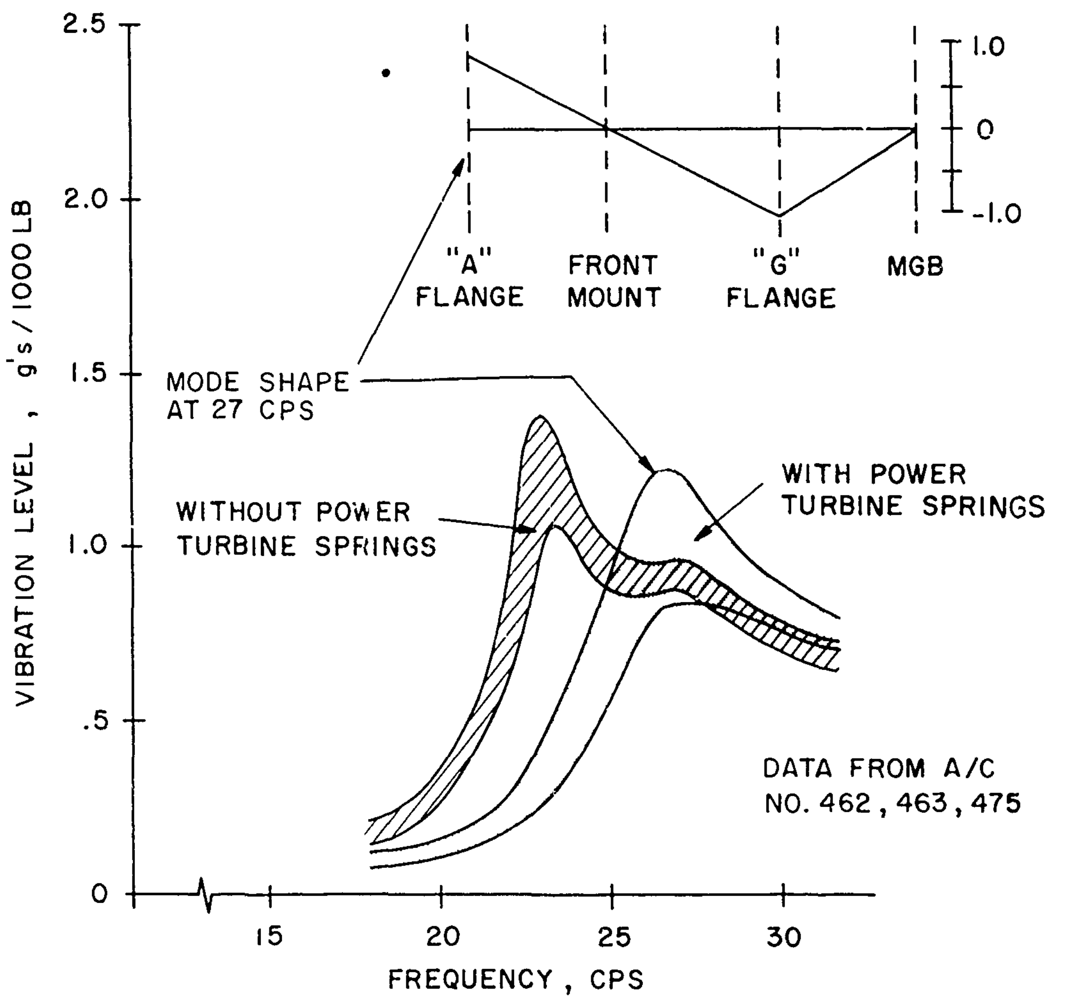


Figure 18. Effect of Power Turbine Spring Restraints.

Problem 5. CH-53E/T64: Engine Longitudinal Mode Excited at 7P

Problem Manifestation

During development testing of the CH-53E propulsion system on the PSTB (propulsion system test bed) the number 2 engine expe ed a drive shaft disengagement from the gearbox that resulted ensive damage to the engine and moderate secondary damage to the tand. This incident occurred in April 1971.

Problem Investigation

For development testing of the three-engine CH-53E propulsion system, the test bed formerly used for the two-engine CH-53A aircraft was modified to accept the additional T-64 engine and the proposed gearbox and rotor system. Installations for the R.H. and the L.H. engines (CH-53E engine numbers 1 and 3) were the same as used in the RH-53D aircraft. Since these engines were already operating satisfactorily in an aircraft installation, a routine set of instrumentation was used to monitor the vibration. Extensive instrumentation was used on the new number 2 engines to measure vibrations in the vertical, lateral, pitch, roll and yaw degrees of freedom. During test operations, none of the instrumentation was indicating unusual vibrations; however, the number 2 engine drive shaft spline became disengaged from the gearbox, with a consequent engine overspeed and damage. Arrangement of the three engines is shown in Figure 19.

Investigation revealed that the engine was longitudinally responsive to the main rotor 7P excitation frequency of 20.6 Hz. It is believed that the relative longitudinal motions were the cause of breakage of the spline shaft retaining clip which allowed disengagement of the spline and subsequent overspeed of the engine. Vibration levels were not measured in the longitudinal direction because there had not been an allowable limit established for this direction.

Natural frequency of the longitudinal mode is governed by the flexibility of the torque tube gimbal ring. Vibration measurements during run-ups of the cold engine indicated that the maximum longitudinal response occurred at 105% N_R (21.7 Hz). Analysis indicated that the gimbal ring longitudinal stiffness was lower than basic data had indicated, and that the mode was probably in fairly close proximity to main rotor 7P at 100% N_R.

The gimbal ring design was analyzed for feasible modifications for making a higher frequency installation. The concept of a lower frequency installation was discarded because it could cause additional wear on the spline due to larger relative motions between engine and gearbox. To expedite testing with a stiffer gimbal ring, a new one was manufactured of steel instead of aluminum as used in the original design. With this stiffer ring installed, rap testing indicated that the fundamental longitudinal frequency was at 47 Hz (15.9P).

Solution

The steel gimbal ring was installed in the PSTB, along with the following redundant features:

1. Ruggedized retention clip was installed.

2. Positive stops were installed so that even without the retention clip, axial motion of the spline shaft is limited and disengagement cannot occur.

The PSTB was operated up to the full spectrum of engine and rotor variation, and vibrations were found to be within specification limits, including the new longitudinal limit imposed by General Electric. Endurance testing was completed without further vibration difficulties. These improvements were used in the aircraft installation and operations were satisfactory. There was no need to impose design improvements on the steel gimbal ring.

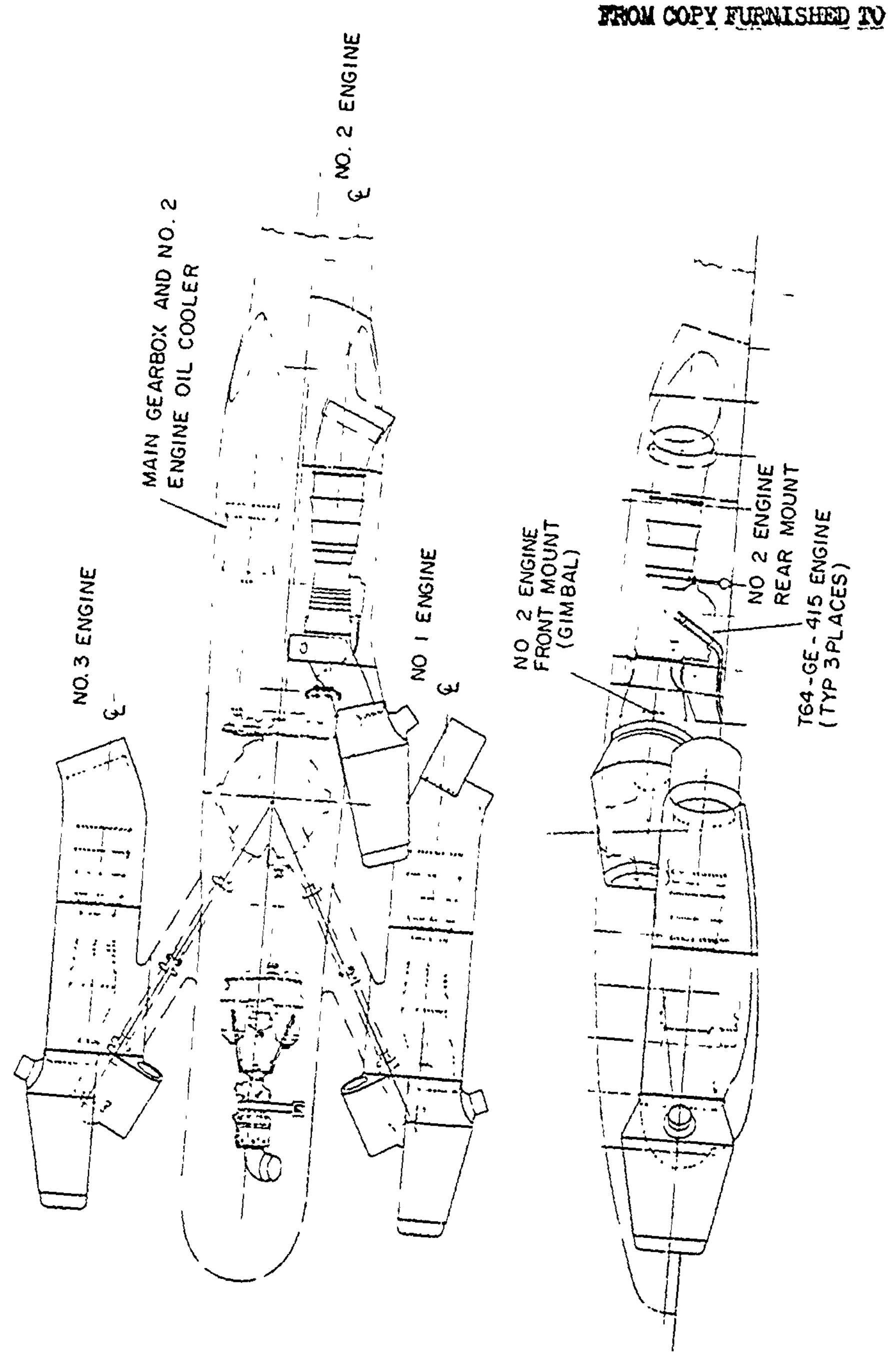


Figure 19. CH-53E Engine Arrangement.

Problem 6. YUH-60A/T-700: Rotor/Drive Train 3rd Torsional Mode Excited at 4P

Problem Manifestation

Operational testing of the YUH-60A propulsion system was performed on the Ground Test Vehicle (GTV). During the early phase of testing, in 1974, large vibrations were experienced on the GTV structure. The vibrations were at the 4P frequency and appeared to be excited by the four-bladed main rotor.

Nature of Problem

The main rotor blade (with hub fixed) had a first edgewise bending mode near 4.6P (1200 cpm) at 100% Nr. When the four blades responded in phase, in a collective mode, the impedance of the hub (reflecting the drive train dynamics) lowered the frequency of the collective mode to 4.1P. The mode combined rigid body lag motion and first edgewise bending mode motion, resulting in little motion across the lag hinge. The result was that the lag damper was not effective, and modal damping was low. The proximity of the mode to 4P caused significant response at 4P (probably excited largely by airframe aerodynamics effects). This rotor response produced 4P vibration of the GTV.

Investigation

Vibration modal analysis of the main rotor blade was performed with an in-house program in which the blade was modeled as 15 masses connected by angular springs representing the stiffness characteristics of the blade. The analysis was uncoupled and only in-plane degrees of freedom were used. Hub impedance effects were included by defining the hub as the first segment of the blade. Flexibility of the drive system was modeled as an angular spring at the shaft centerline. A sketch of the analytical model is shown in Figure 20.

The drive system impedance at the rotor shaft was calculated using a Holzer-type analysis, which used drive system torsional moments of inertia and torsional springs between components. A frequency is assumed and the calculation determines the impedance required at the upper end of the main rotor shaft to give the assumed natural frequency. The system natural frequency was found by finding the frequency at which the assumed blade root impedance matched the drive system impedance. The impedance/natural frequency curves for the drive system and the blade-hub system are shown in Figure 21. The 3rd torsional natural frequency of the system was found to be 4.1P, which corresponds very closely to the 4P vibration measured during test.

A sketch of the displacement shape for the 4.1P mode is shown in Figure 22. The shape indicates that the hub motion just about matches the computed blade deflection at the hinge. Therefore, there is very little angular motion at the lag hinge. This was characteristic of the test data as well. This absence of lag motion is a contributing factor to the problem, for without lag motion across the hinge the lag dampers are ineffective and cannot damp out any response of the mode.

The analytical model was used to evaluate weight and stiffness parameter variations, with the following results:

1. 10-lb weight inboard on blade lowered frequency from 4.1 to 3.95P.

2. Increase in spindle stiffness varied mode from 4.1 to 4.2P.

3. Increase in edgewise blade stiffness raised mode from 4.1 to 4.4P (The first edgewise blade-alone mode went from 4.6 to 5.3P).

Experience with this problem motivated the development of a fully coupled rotor/drive system dynamic analysis capability at Sikorsky.

Solution

The solution chosen was to add graphite stiffening strips along the blade trailing edge. These blades were tested on the bench and on the whirl stand to confirm the increase in nonrotating and rotating edgewise natural frequency. They then were tested on the GTV. No problems were encountered with the modified blades, and the increase in edgewise stiffness became the final solution.

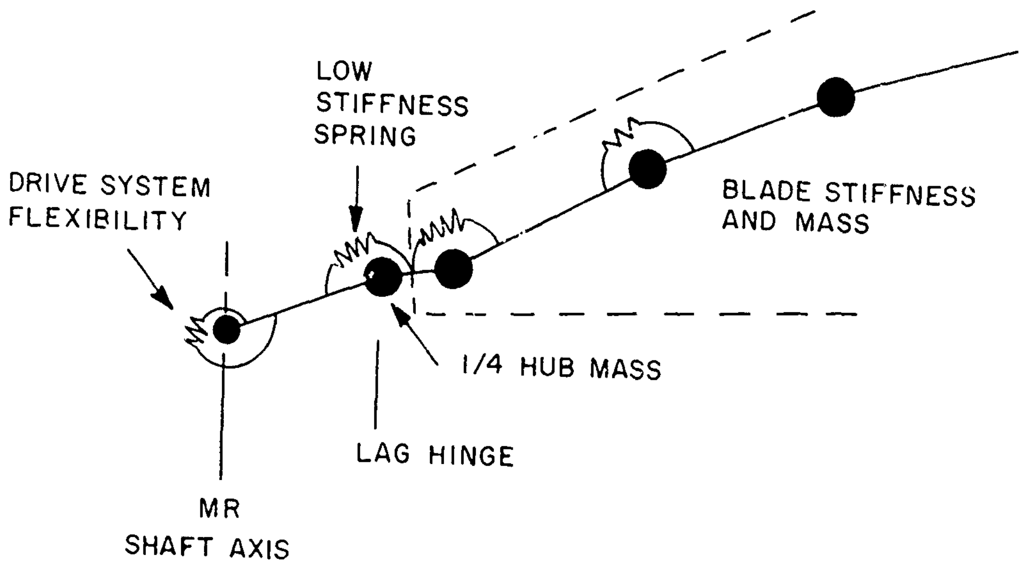


Figure 20. Rotor Analytical Model.

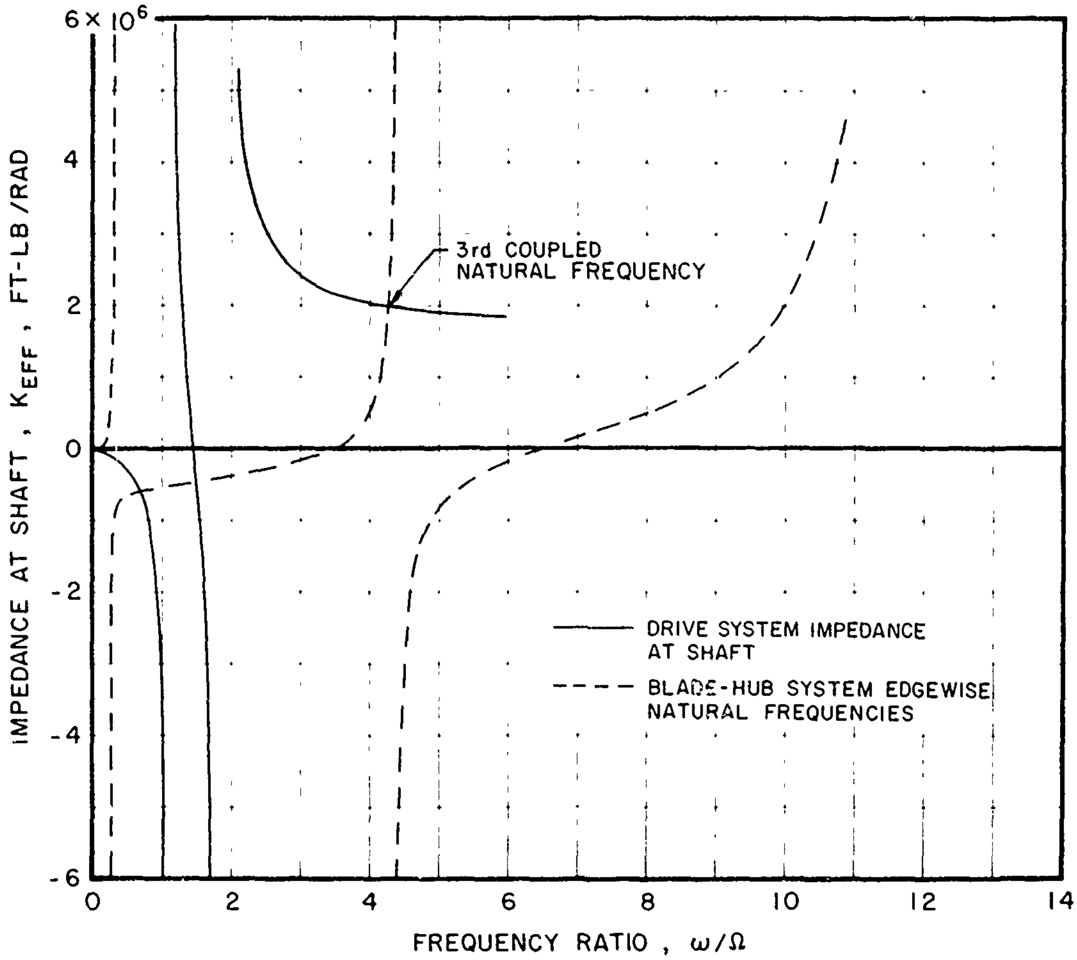


Figure 21. Natural Frequencies of Rotor/Drive System.

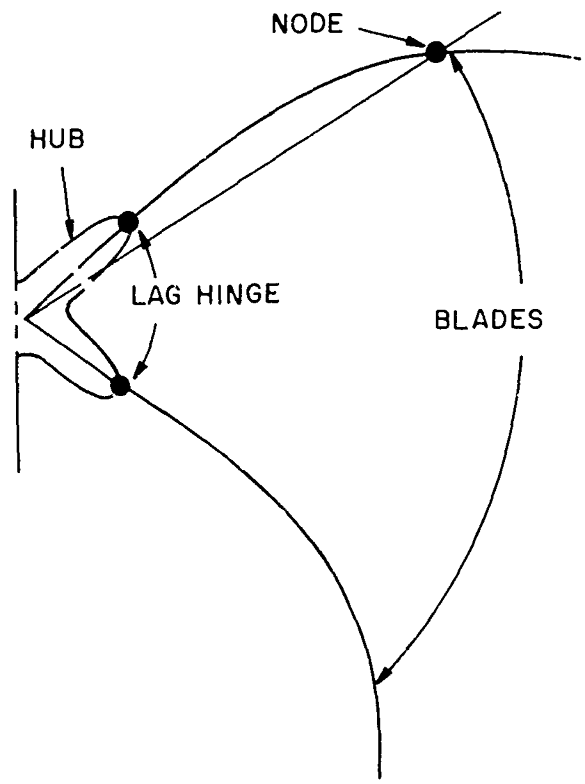


Figure 22. 3rd Torsional Mode: Detail of Blade Motion.

FORCED VIBRATION PROBLEMS, MECHANICAL IMBALANCE EXCITATION

Problem 7. SH-3A/T-58: Engine Torque Tube Bending Mode Excited at N_F

Problem Manifestation

For nearly three years the development and production of the SH-3A aircraft proceeded without serious engine (T-58) installation problems. Then, rather suddenly (in June 1962), an almost exponential increase in the number of engine installation discrepancies was reported (Figure 23), the majority of which were fatigue cracks in the power turbine bearing housing flange.

Nature of Problem

Measurements taken at this location (originally not a specified measurement location) revealed 1 X N_f vibration levels in excess of 9 mils D.A. This was well above the 3-mil maximum specified at the standard locations. A subsequent N_f frequency response sweep revealed an engine resonance occurring in the operating range of 1 X power turbine speed 314 Hz (Figure 24).

The engine installation is shown in Figure 25. The resonant mode shape is shown in Figure 26. It is seen to involve the bearing housing and the torque tube, but the shell of the engine appeared to be isolated from it.

Problem Investigation

To determine the cause behind the sudden increase in the failure rate, the effects of design changes incorporated during the previous 6-month period were investigated. The critical change was found to be switching the gimbal bushing material from nylon to laminated steel-polyurethane. This had been done in order to reduce the excessive wear experienced on the nylon bearings.

Subsequent tests showed high vibration levels (at the bearing housing) for both the polyurethane and the nylon bushings when they were new. However, when worn nylon bushings (.030" slop) were used, the resonance was removed from the operating region and the vibration levels were reduced to within 4 mils (Figure 27). It was concluded that the nylon bushings had acted as a fuse to contain the vibration. The excessive response and resonance, which went with the new nylon bushings, wore the latter down, causing enough slop to then detune the installation and lower the response. With the introduction of the polyurethane bushings, the slop was eliminated and the original resonance was maintained.

Solution

Since nylon could not be reinstated into production and since all attempts at variability and balance were exhausted, it was decided to lower the resonance below the operating range. It was estimated that the effective mass of the resonant mode was approximately 40 lb. Therefore, an added mass of 12½ lb was considered sufficient to lower the mode to approximately 85% N_f. A 12½-lb weight was clamped to the torque tube at the number 5 bearing housing junction (Figure 28).

As shown in Figure 29, the weight lowered the resonance to 85% N_f which resulted in vibration levels within 3 mils at the bearing housing throughout the operating range.

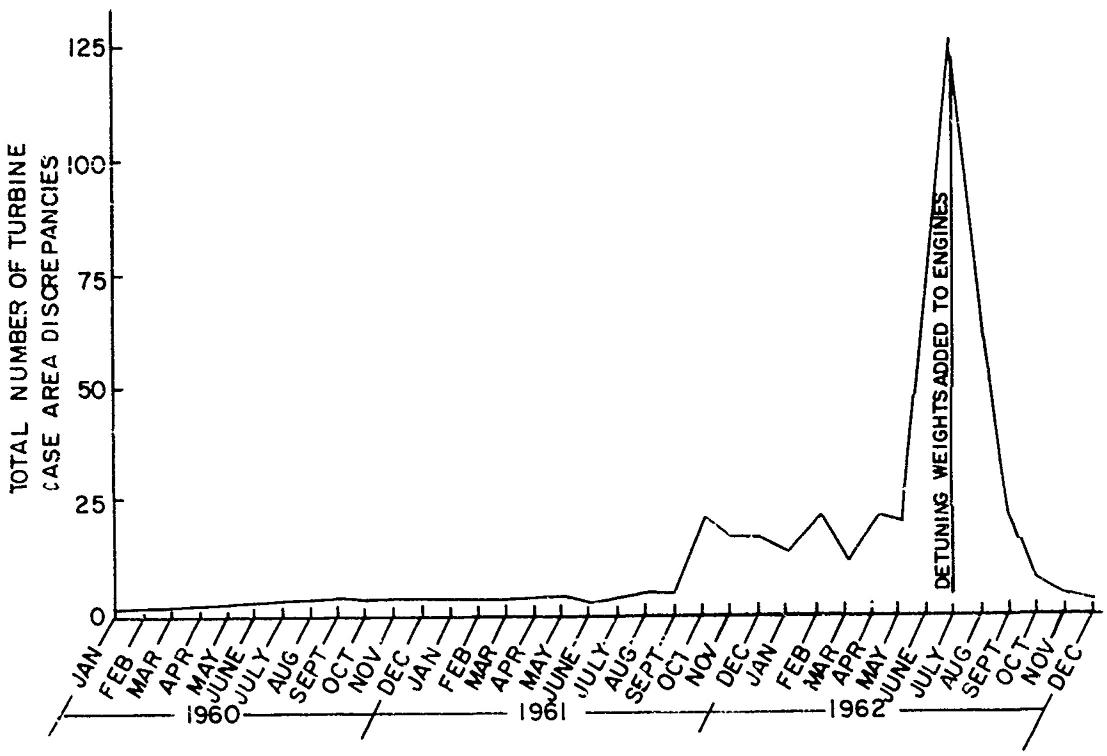


Figure 23. History of Power Turbine Case Area Discrepancies.

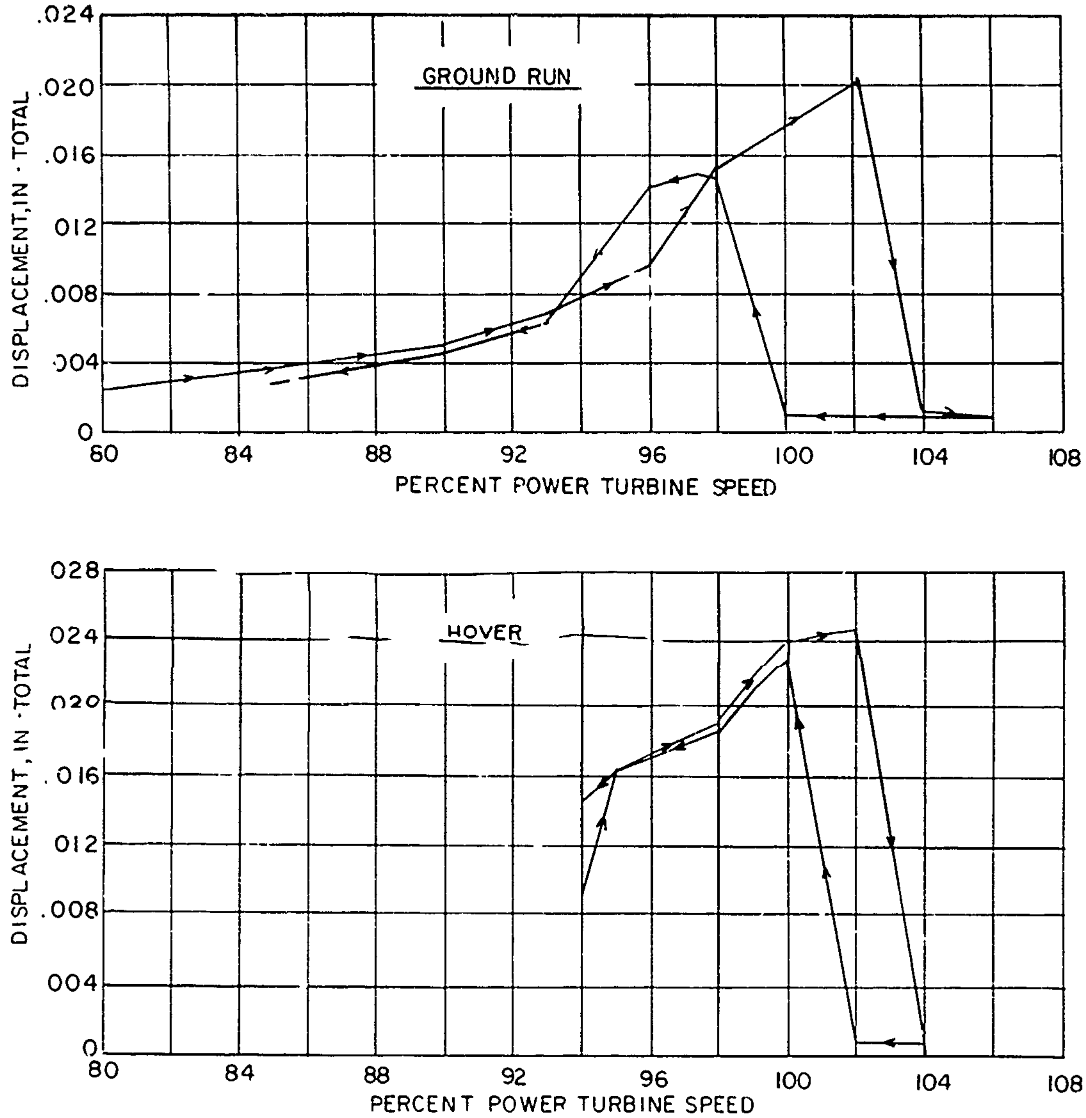


Figure 24. Power Turbine Bearing Housing Lateral Vibration.

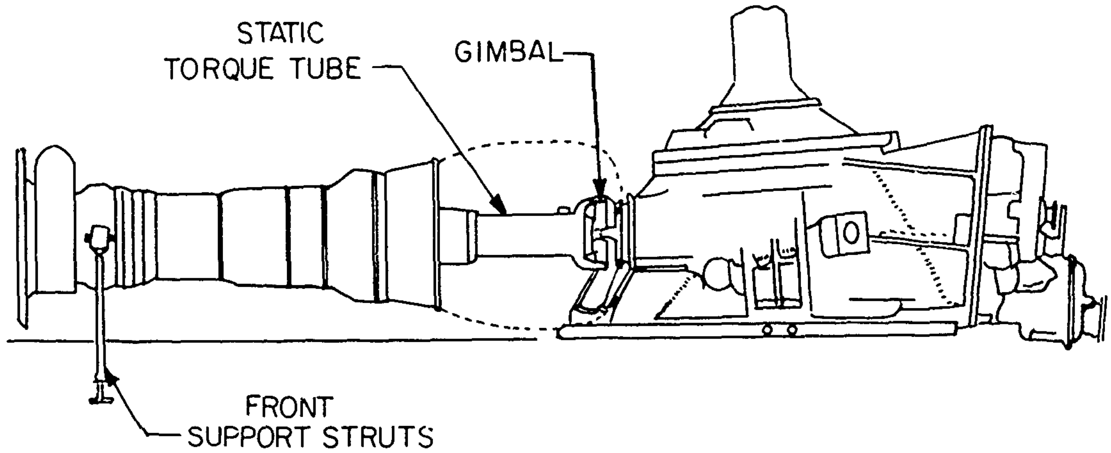


Figure 25. SH-3A Engine Installation.

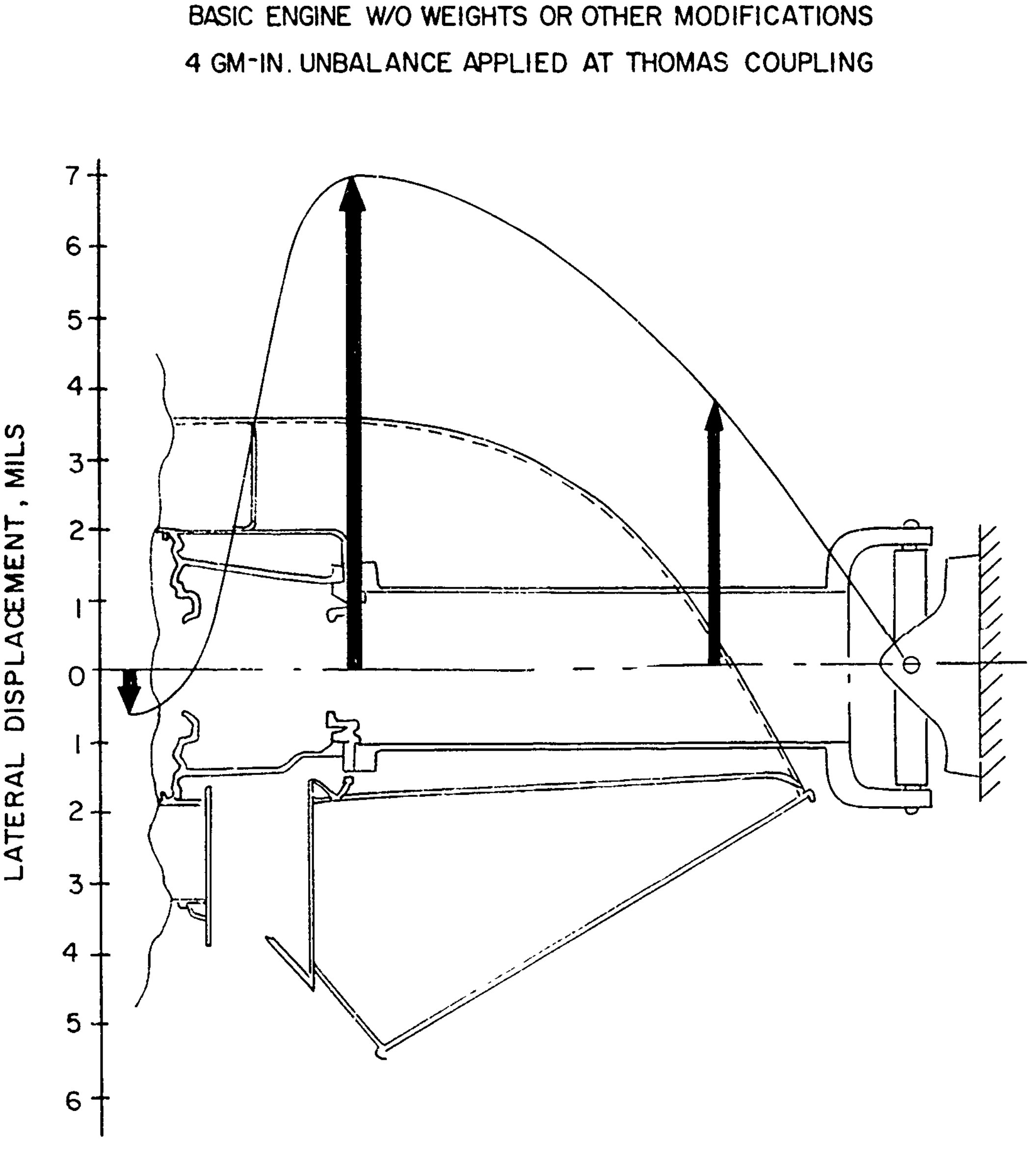


Figure 26. Torque Tube Response Shape at 1 X N_F.

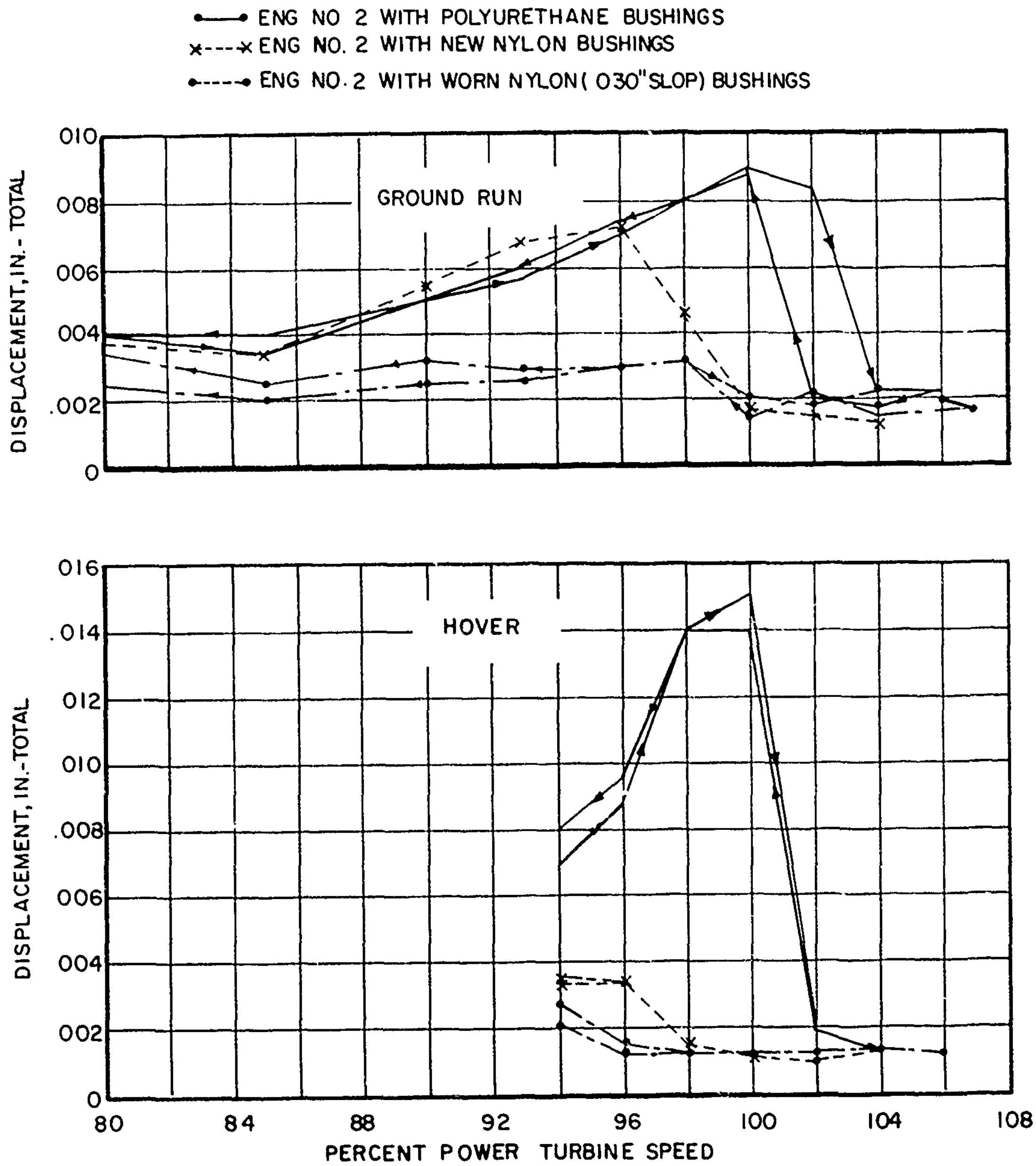


Figure 27. Comparison of Polyurethane and Nylon Torque Tube Bushings.

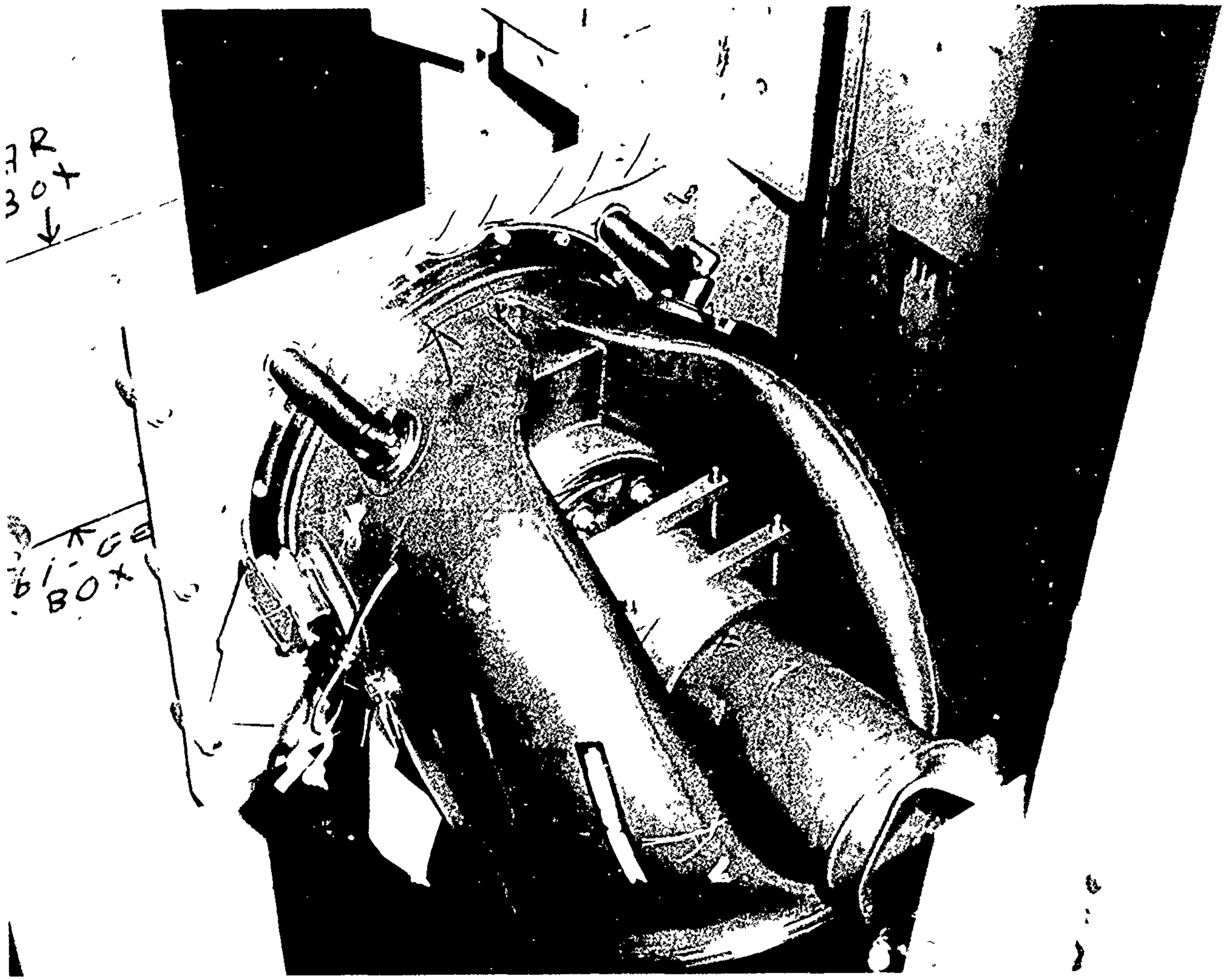


Figure 28. Twelve and One-Half Pound Weight Clamped to Torque Tube.

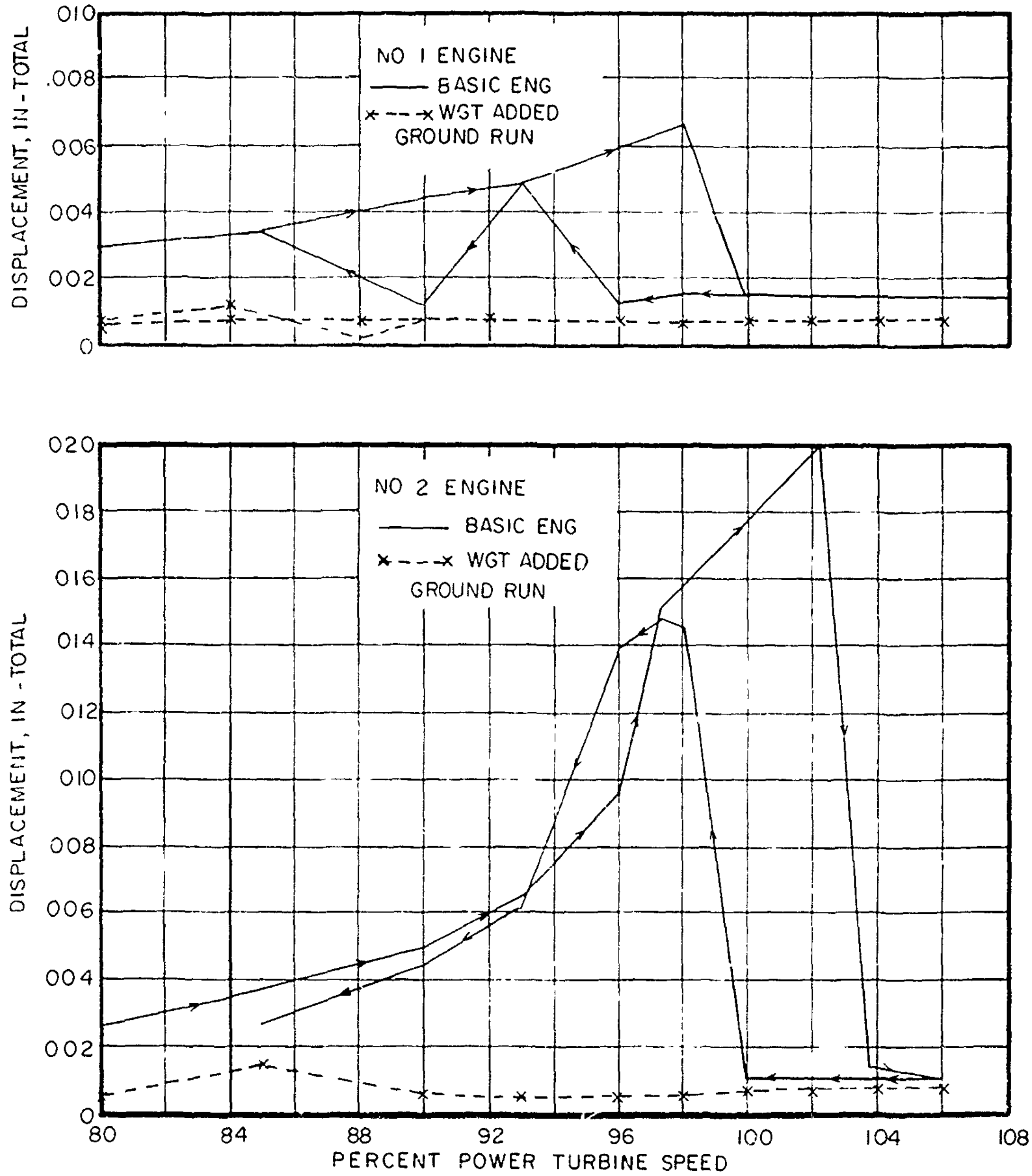


Figure 29. Engine Vibration - Effect of Adding 12½-Pound Weight to Torque Tube.

Problem 8. S-58T/PT6T-3: Engine Drive Shaft Critical Speed

Problem Manifestation

A drive shaft critical speed investigation was conducted on a tiedown Sikorsky model S-58T helicopter. This was done as part of the FAA requirement for substantiation of dynamic components new to the S-58T turbine engine conversion. During testing in 1970, excessive vibration was measured at the engine drive shaft at N_{shaft}(110 Hz).

Nature of the Problem

The S-58T helicopter is a S-58 model with a power plant conversion from piston to turboshaft engine. As part of the substantiation of the dynamic components new to the S-58T, the absence of critical speeds throughout the operating speed range of the aircraft drive shafts was demonstrated. In addition, the fundamental frequencies of the drive shafts, angle gearbox, and blower were determined and a critical speed investigation was carried out on the angle gearbox and blower. These components are indicated in the sketch, Figure 30, along with the indicated pickup locations. The engine drive shaft is a new design (manufactured both in aluminum and steel), whereas the angle drive shaft is the same as used with the S-58 installation except that it has a changed coupling for the S-58T installation. During rotor system runups, with steel engine drive shaft installed, there was a speed-dependent vibration response which reached 6.7 g at 104% rotor speed.

Problem Investigation

Fundamental vibratory frequency responses were determined on each component by using transient response techniques. The engine drive shaft transverse bending frequencies are presented below with pertinent operating range data:

		steel	aluminum
bending frequency	Hz	146	165
	CPM	8760	9900
operating speed	RPM	(5870 to 6860)	
	% NR	(89% to 104%)	
(bending f/operating f)$_{min.}$	%	128	144

Frequencies of the angle drive shafts, angle gearbox, and blower are not pertinent to this report and are therefore not presented.

For the critical speed investigation, pickup responses were recorded while the rotor speed was varied from zero to the rotor red-line speed of 104% NR (258 RPM). Special attention was paid to the speed range from 89 to 102% NR which is the normal operating range. With the aluminum engine drive shaft installed, the overall induced vibration levels remained low throughout the operating speed range. The steel engine drive shaft installation, however, had a speed-dependent vibration magnitude with frequency equal to the engine drive shaft speed. The vibration magnitude reached 6.7g at 104% rotor speed at pickup 3. The source of vibration was verified as the drive shaft by running for a short time with the oil cooler blower disconnected (the blower runs at the same speed as the drive shaft) and observing that the vibration level did not change.

In order to determine the nature of the induced vibration in the steel drive shaft installation (i.e., due to simple unbalance or due to operation close to a critical speed), the following analysis was carried out. Data points from the steel drive shaft, pickup 3, are presented in Figure 31. The solid line is an exponential least-squares fit through these data points. If the vibration were induced by a simple unbalance, the amplitude would be proportional to the square of the rotational speed. This "mass unbalance" line is shown in Figure 31 where it is drawn through an arbitrary set point. On the other hand, if the vibration were induced by approaching a critical speed, the curve shape would be that of the classical forced vibration response of a single-degree-of-freedom system which peaks at the resonant frequency. The equation for this curve is shown in Figure 31, where the natural frequency used was 145 Hz and the damping coefficient was 0.038 as measured during the transient response tests. This curve was adjusted to pass through the same set point as the mass unbalance line and is also plotted in Figure 31. The close correlation of the forced response curve to the measured curve indicates that the vibration is more likely to be of the shaft critical type, and the critical speed could occur at 145 Hz (main rotor 133% N_R).

Solution of Problem

Although the steel engine drive shaft would not be operating near its critical frequency, it was decided to use the aluminum drive shaft because of the lower vibration levels induced in its installation.

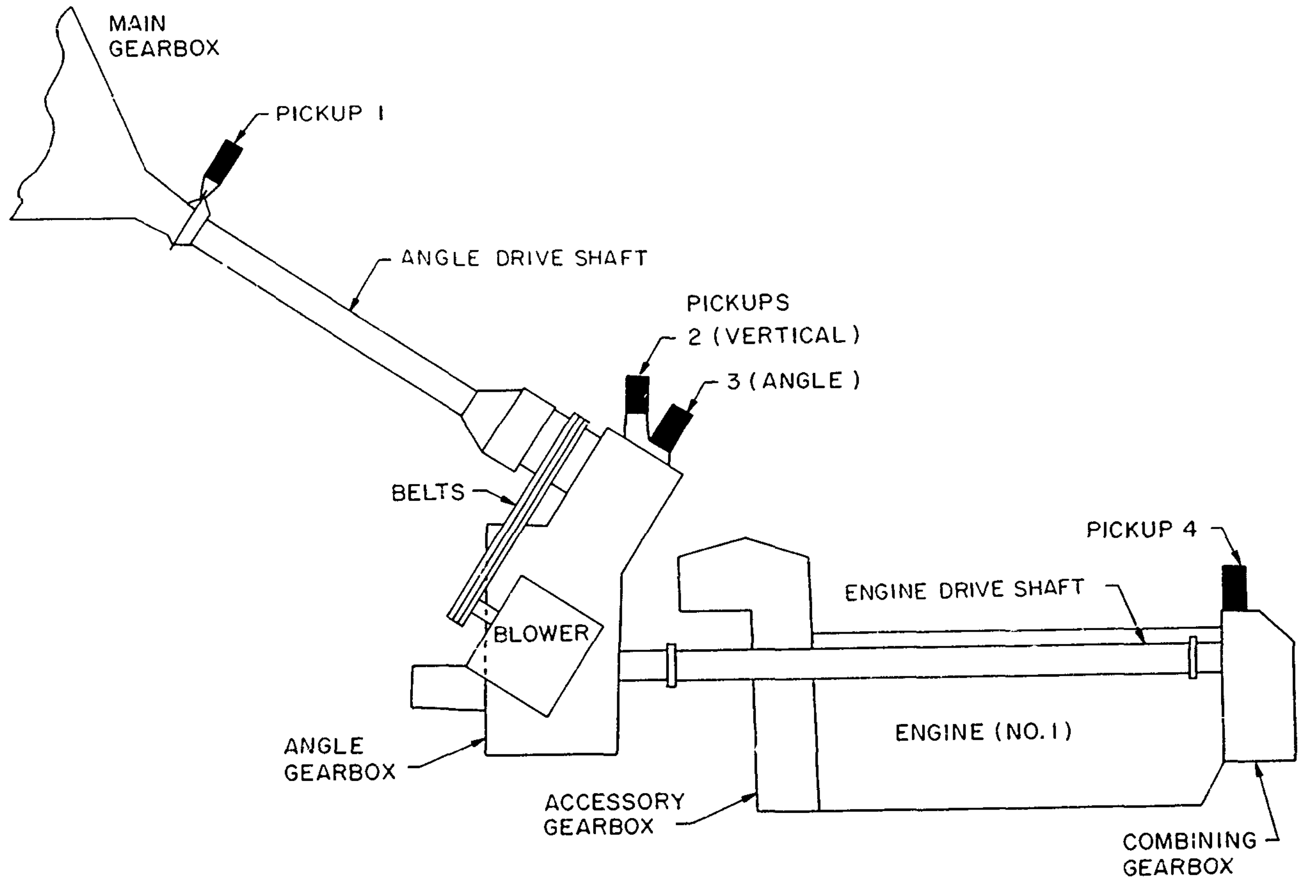


Figure 30. Velocity Pickup Locations - S58T Tiedown Aircraft.

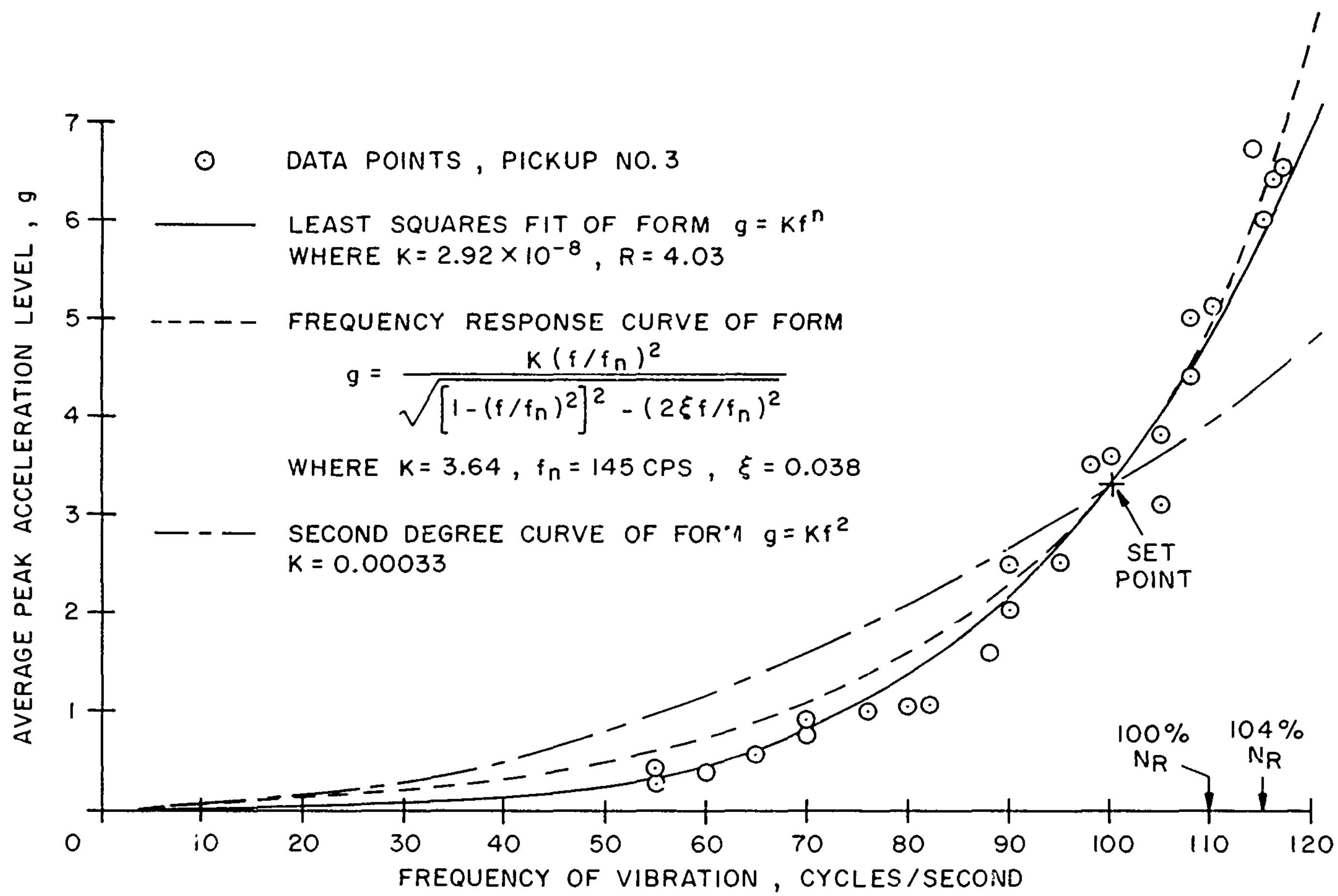


Figure 31. Vibration Response Curves.

Problem 9. NSH-3A/YT58-GE-16: Engine Drive Shaft/Torquemeter Critical Speed

Description of Problem

A roller gearbox was developed under a U.S. Army contract during the period 1968 to 1974. During tiedown tests of the new gearbox propulsion system in a NSH-3A aircraft in 1971, large vibration values were measured on the static torque tube, plus evidence was noted of torque tube/transmission gimbal mount rubber damage after 28 test hours.

Part of the roller gearbox contract was to develop a phase-line torquemeter. This torquemeter sensing element was incorporated in the engine/gearbox drive shaft which is located within the static torque tube. This is a different kind of torque-measuring device than used before. Because of its high operating speed, up to 22,190 RPM, and because of its proximity to the gimbal mount, it was suspected of being the source of unbalance excitation. A view of the drive shaft in the back-to-back test setup is shown in Figure 32, and a picture of the drive shaft is shown in Figure 33. The torquemeter system is shown schematically in Figure 34, and an engine installation view is shown in Figure 35. As shown in Figure 32, the engine output end of the drive shaft has attached to it an outer shaft having three equally spaced pole pieces machined into its free end. Attached to the gearbox end of the drive shaft is another outer shaft also with three equally spaced pole pieces which intermesh with the pole pieces on the input end. As the engine drive shaft twists under the influence of shaft torque, the spacing between the pole pieces varies. The spacing change is detected by a magnetic pickoff device attached to the fixed torque tube. The pickoff signal is conditioned and then displayed on the instrument panel as a percentage of transmitted torque. The drive shaft is designed for a nominal windup of $1.56^{o} \pm .06^{o}$ when transmitting 100 percent torque (6214 in-lb). The maximum windup at 130 percent torque is 2.03^{o}.

In the design of the torquemeter/drive shaft assembly, the natural frequency, or critical speed of the assembly, had to be determined to ensure that resonance would not occur within the operating range of 0 to 22,190 rpm, which corresponds to a 117-percent autorotative red-line speed. The critical speeds were determined by computer analysis. The drive shaft outside diameter and wall thickness parameters were varied within the constraints of the torque tube diameter. The drive shaft length, between the flexible steel laminated coupling and crowned spline coupling, was predetermined by the requirement to accommodate the engine exhaust casing. Figure 36 shows the drive shaft layout and the normalized shaft deflection for the first critical speed.

Problem Investigation

As development testing was continued, attempts to obtain useful data from the torque unit were not successful because torsional stiffness was too great to provide the necessary displacement sensitivity. Sensitivity increases produced by reducing stiffness would have reduced the shaft critical mode frequency from 26,669 CPM to a value within the operational range. A stabilizing member was used at the aft end of the longer outer shaft to provide support by bearing on the inner shaft. This stabilizing element was later modified to provide clearance from the inner shaft because it was thought that drag/friction forces might be affecting the relative torsional displacements. In operation at high RPM, centrifugal force on the pole pieces caused them to be displaced outward. The local inertia/elastic properties of the pole pieces had not been precisely controlled. Therefore the slightly different outward displacements caused the outer shaft to become unbalanced. The shaft vibration was made worse because the stabilizer element clearance allowed outer shaft bending deflection to increase the unbalance.

The engine vibration pickups mounted at the General Electric specified locations indicated that the amplitudes of vibration were within the manufacturer's required limits of 3 mils double amplitude in steady state and 5 mils double amplitude transient (2 seconds maximum). Of the six pickups on each engine, the highest in amplitude are shown in the graphs of Figure 37. The G.E. limits are also shown for the engine crotch location. Number 1 engine vibration levels were less than those of Number 2 engine.

The vibration pickups located on the engine torque tube are not engine pickups and are therefore not subject to the GE limits. Plots for these pickups are presented in Figure 37 where some evidence of a resonance is seen at 90-percent rotor speed. No limits have been determined for these locations; however, the vibration levels observed there are undoubtedly the major contribution to the gimbal mount deterioration.

When the outer shafts with the integral pole pieces were removed from the drive shaft, the torque tube vibrations were reduced to normal levels.

Problem Solution

The outer shafts with integral pole pieces were removed from the drive shaft, and roller gearbox development was continued. Further development work on the torquemeter system was discontinued.

It was found that adequate power measurement and power matching of engines was possible because main and tail rotor torques were used as a measure of power, and the engines were matched by using T5 and Ng.

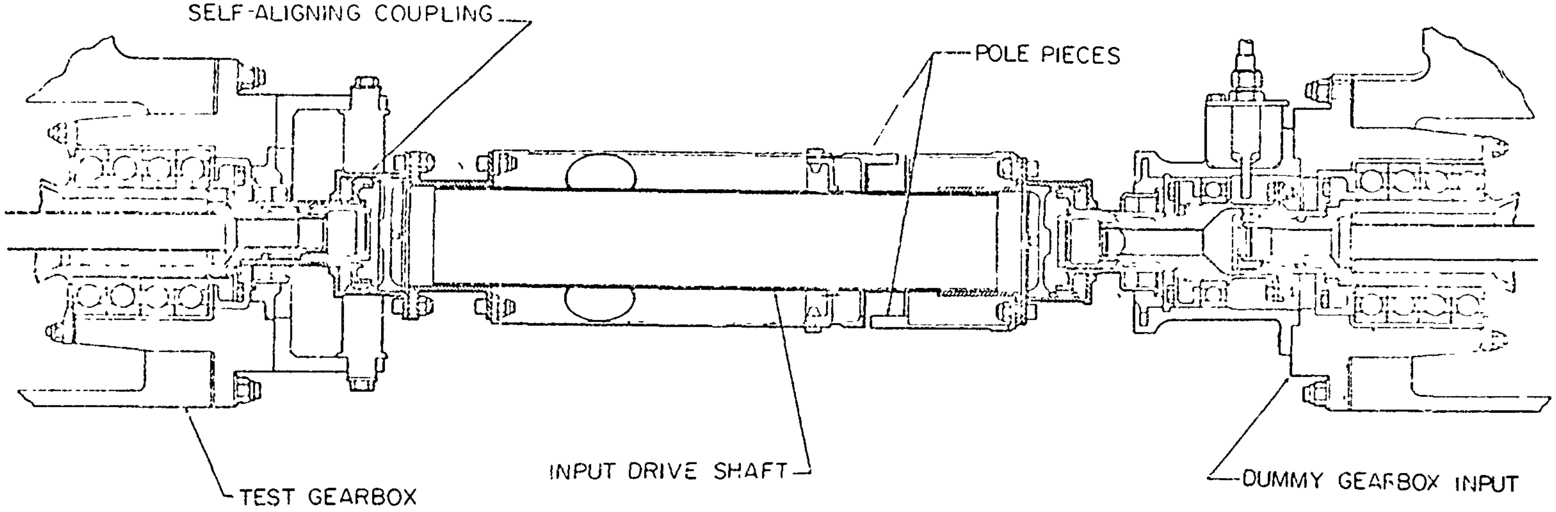


Figure 32. Drive Shaft Installation - Regenerative Test Stand.

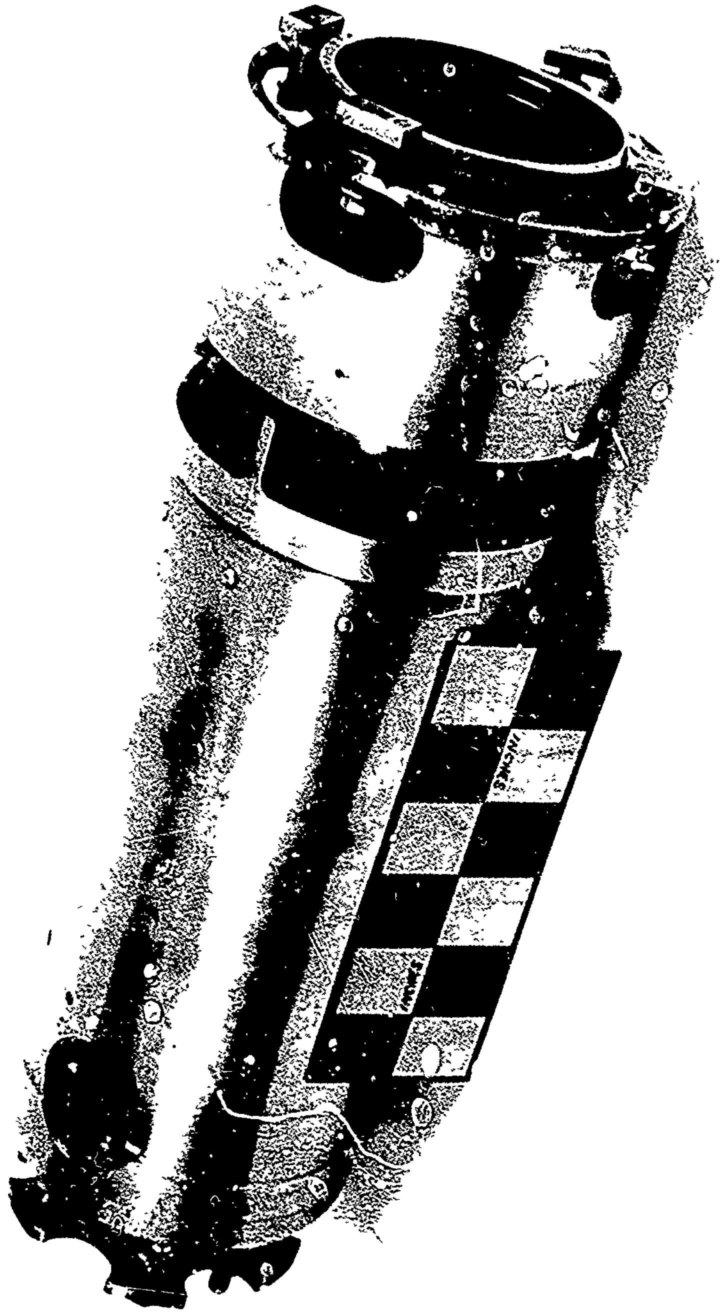


Figure 33. Engine Drive Shaft with Pole Pieces.

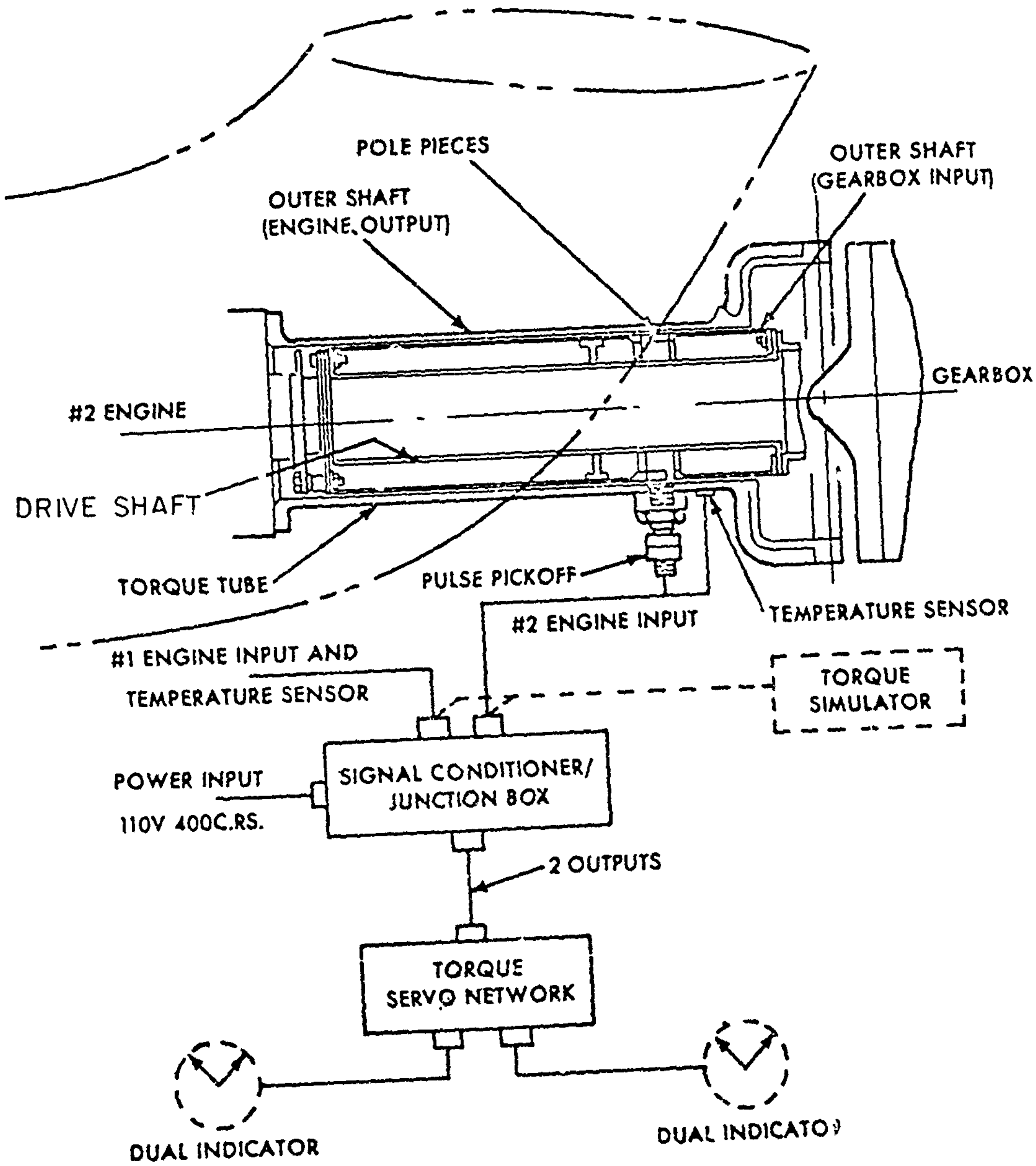


Figure 34. Engine Torquemeter System.

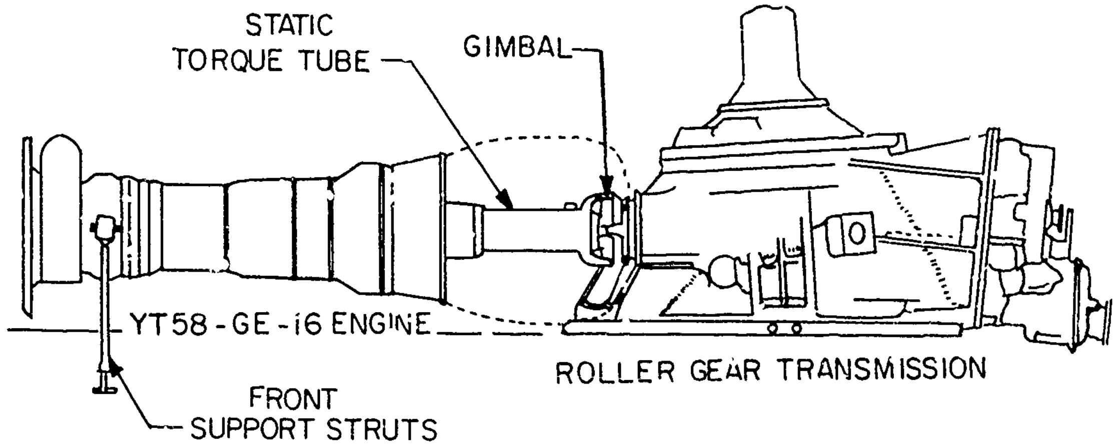


Figure 35. Engine Installation Drawing

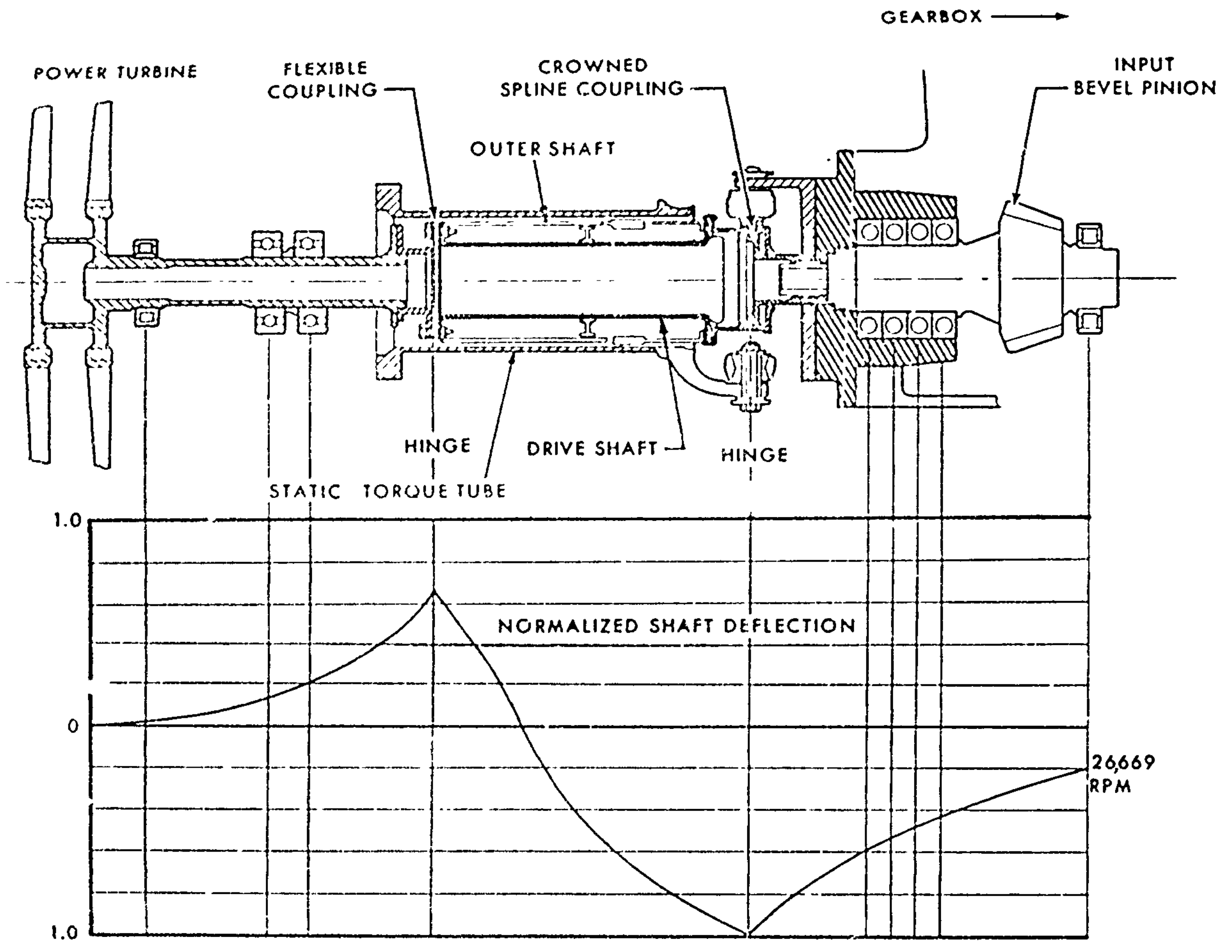


Figure 36. Input Drive Shaft Layout - Critical Speed Analysis.

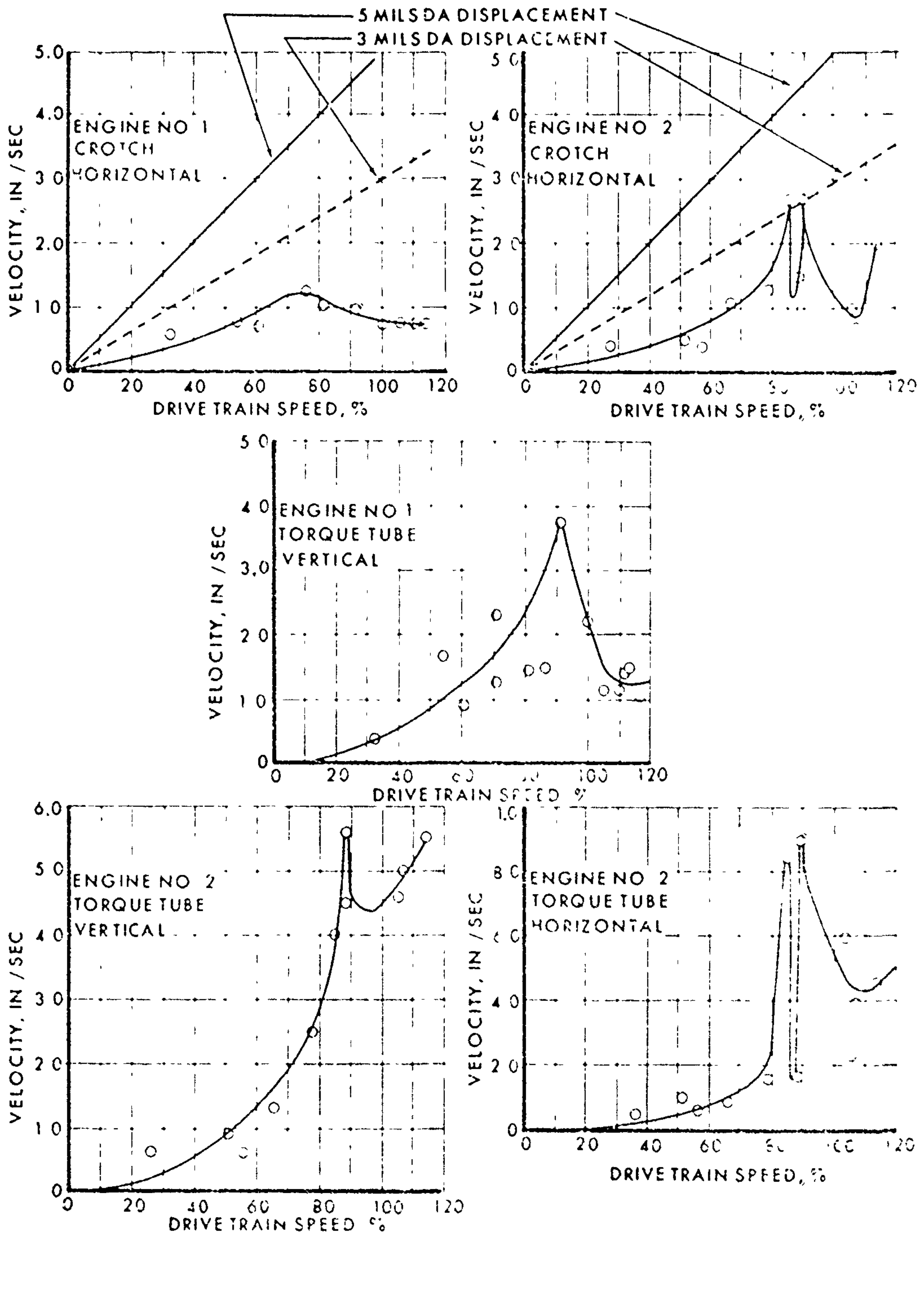


Figure 37. Engine Vibration Graphs.

FORCED VIBRATION PROBLEM, OTHER EXCITATION

Problem 10. YUH-60A/T700: IPS Shaft Torsional Mode Excited by Rotating Starter Jaw

Problem Manifestation

The YUH-60A helicopters were powered by a pair of T700GE engines. An integral particle separator (IPS) was one of the items of equipment driven by the accessory gearbox of each engine. The IPS is a blower which is used to scavange the unclear air inertially separated from the engine inlet airflow. In 1975, during the basic engineering development program, many torsional failures were detected in the IPS rotor shaft. Failures were uncovered during routine inspections and occurred during operational exposure times varying from 32 to 247 hours, and with the number of starts from 27 up through 407. A typical failed shaft is shown in Figure 38. Failures are tabulated in Table 3.

Problem Investigation

The IPS blower is driven at 30,000 RPM, at 100% N_R. To reduce the possibility of damage to the accessory gearbox in the event the blower suffered damage, the blower shaft incorporated a locally reduced cross section area (shear weak link) with an endurance limit torque of 48 foot-pounds. All the shaft failures occurred at this area. GE reviewed failures in the UH-60A/T700 installation and made comparison with blowers in other aircraft installations using T700. They found that failures had occurred only in the UH-60A aircraft, and that the source of large torque transients appeared to be in the starter unit. The starter is a pneumatic unit manufactured by Bendix which operates from a ground gas generator unit or from bleed air from a T700 engine. Gearbox transient torques could be caused by starter slip/skip/chatter of the teeth during the initial engagement of the drive jaw to the driven jaw. Figure 39 presents a view of the connecting jaws.

Qualification and acceptance testing of the starter at the Bendix plant had not shown large torque transients. Similarly, early measurements on a starter unit in the Ground Test Vehicle (GTV) had not revealed large torque transients. Improved higher frequency instrumentation was then used on the IPS shaft and starter jaws, both at Bendix and on the GTV, and both sets of data revealed that large transient impact torques of approximately .015-second duration occurred on starter engagement.

Nature of Problem

Starting torques on the IPS shaft were measured to have spikes as large as 76 foot-pounds, as compared to the 48-foot-pound endurance limit. The IPS fundamental torsional frequency was determined by GE to be 180 Hz. Apparently it was the response of this mode to the torque spikes that was generating a sufficient number of cycles and stress magnitude to cause the torsional failures.

Conditions supporting large transient torques during starting engagements are:

1. Fast rate of "spin-up" of starter jaw

2. Jaw misalignments due to shaft bending, thus causing poor mating of jaw teeth

3. Attempting starter engagement while engine is still rotating at a high speed

In order to continue the flight aircraft in the Government Competition Test (GCT) phase, the following temporary changes were imposed:

1. Operational restriction was placed on the aircraft to prevent starter reengagements at high speeds of the accessory gearbox (high gas generator speeds).

2. Bendix reset the starter control valve to a spin-up (pressure rise rate) as low as possible, consistent with starter performance.

3. GE doubled the IPS shaft shear area "weak link" to produce a twofold increase in endurance strength.

4. Adding local jaw guide to reduce shaft bending on the starter side was considered to be a good fix; however, it was not possible to include it before committing the aircraft to GCT.

Results of the above temporary fixes were that there were no more IPS shaft failures during GCT and in maturity testing up to the end of 1977.

For the production UH60A, the starter will use an overrunning clutch which has demonstrated low impact torques. This clutch has been used in other aircraft and did not cause IPS shaft failures. Use of this clutch will permit reengagements up to starter cutout speed, and pressure rise rate will not have to be reduced. The doubled shear area will be retained in the IPS shaft.

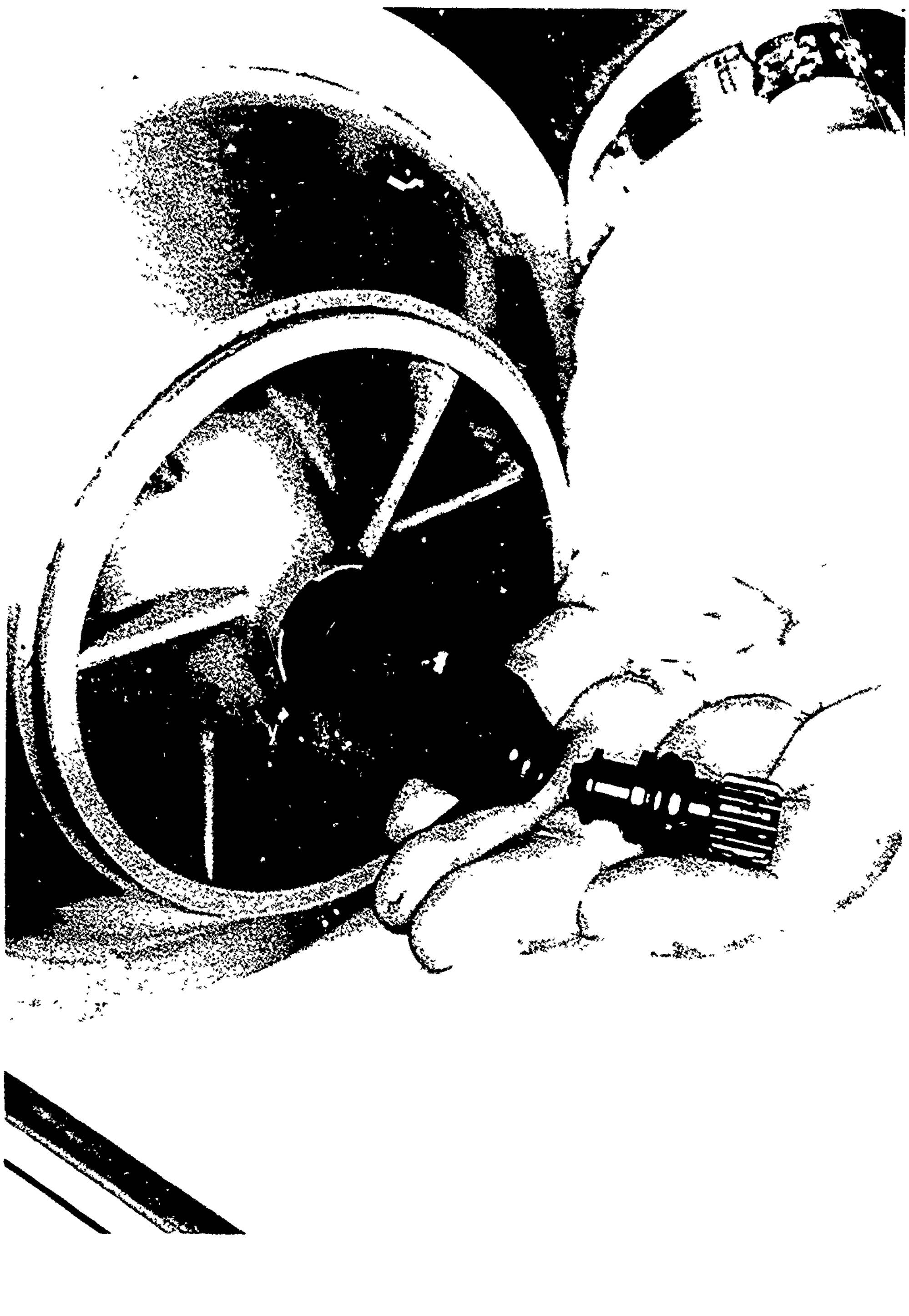


Figure 38. Typical Failed IPS Rotor Shaft.

Table 3. IPS Blower Failures.

	ENGINE	BLOWER* HOURS	BLOWER S/N	NO OF STARTS	DATE	FAILURE MODE
1.	XT 105	106	TAS-17	--	5/1/75	Torsional Shear
2.	XT 111	214.7	TAS-02	65+	5/19/75	Torsional Fatigue
3.	YT 218	147.8	TAS-49	228	9/10/75	Torsional Fatigue
4.	YT 218	126.8	TAS-46	150	10/14/75	Torsional Fatigue
5.	YT 214	149.4	TAS-18	175	10/28/75	Torsional Fatigue
6.	XT 108	246.7+	TAS-09	181+	10/29/75	Torsional Fatigue
7.	YT 201	218.2	TAS-06	407	10/29/75	Torsional Fatigue (Star Pattern)
8.	YT 202	211.4	TAS-08	375	10/29/75	Under Investigation
9.	XT 108	32.10	TAS-15	27	11/7/75	Under Investigation

* Time on blower when failure discovered.

Figure 39. Starter Drive Jaw and Accessory Gearbox Driven Jaw.

SELF-EXCITED VIBRATION PROBLEMS

Problem 11. SH-34H/T-58: Engine Load Sharing Oscillation

Background

In the late 1950's, Sikorsky (under Navy contract) modified two SH-34H (HSS-1F) helicopters by replacing the reciprocating engines with two General Electric T-58 free turbine engines. One of these helicopters was used for tiedown ground testing and the other for flight testing.

The purpose of the program was to provide a flight test vehicle for testing of a dual free turbine propulsion system. This was the first time two free turbine type engines had been installed in a helicopter propulsion system. The problem discussed in this writeup is without documentation at Sikorsky and therefore is based on the writer's recall only.

Manifestation of Problem

During the ground and flight test program it was observed that engine load sharing was unstable. It was found that under most operating conditions it was not possible to maintain 50/50% load sharing between the two engines. As a result of this poor load sharing, each time the power required changed due to collective pitch change or wind gust load, sharing would be upset and would begin to oscillate at low frequency ($<$ 1 Hz). Considerable pilot effort would be required to readjust the engine speed selectors to regain a reasonable load sharing. A plot of typical torque levels for good and poor load sharing is shown on Figure 40.

Cause of Problem and Solution

The hydromechanical fuel controls used on the T-58 engine operate on a power-demand basis. A fly-ball type governor is used to monitor free turbine speed (N_f) and has a range of adjustment of approximately 95 to 105%. The N_f governor set point is controlled by the pilot using the speed selector control. The N_f governor in these early fuel controls had a droop curve of approximately 2%. This small droop was considered desirable because rotor speed changes due to changes of power required would be minimized; i.e., a change of power required of minimum to maximum would result in only 2% reduction in rotor speed. However, the governor droop serves a second purpose: it provides load sharing between the engines when multiple engines are used to drive a common gearbox/rotor system. This early experience demonstrated that 2% was too shallow a droop curve to provide satisfactory load sharing and load sharing stability. The solution for this problem was to increase the N_f governor droop from 2% to 6-8%.

Figure 41 shows an example of how governor droop affects load sharing stability. The example assumes that both engines have 8% droop controls and that the pilot has adjusted the speed selector to provide 100% rotor speed while demanding 50% total power. Assuming stable conditions, no collective stick motion or wind gusts, the engines are operating at point "C" on the droop line A-B. Assume that engine number 1, due to a wind gust, increases its power output from 50% to 55%. Engine number 1 moves down the droop line to point "D". Since the total power required has not changed, engine number 2 is required to produce 45% and moves to point "E" on the droop curve. At this point it can be seen that the number 1 engine governor is demanding a speed of 99% whereas engine number 2 is demanding a speed of 101%. Since both engines are driving a common gearbox/rotor system the actual free turbine speed of each engine is identical at 100% Nf. The result is that the number 1 engine governor is sensing an overspeed condition (demand speed 99%, actual speed 100%) and the number 2 engine is sensing an underspeed condition (demand speed 101%, actual speed 100%). Based on these error signals, number 1 engine reduces fuel flow and number 2 engine increases fuel flow, thereby bringing both engines back to a 50/50% load sharing condition.

Following the above logic it is obvious that increasing the droop will provide increased load stability.

Experience subsequent to these early multi-turbine engine tests has shown that N_f governor droop in the order of 3% provides excellent load stability. Engines developed subsequent to the T-58 such as the GE T-64 and Pratt & Whitney JFTD12A have incorporated an additional feature on the fuel control which accepts a collective stick input. This collective stick input (collective bias) is used to reset the droop line upward as collective pitch is increased and downward as collective pitch is reduced. By so doing, the droop built into the fuel control is compensated for as large changes in power required occur, and the rotor speed remains constant as these power required changes occur.

The collective bias feature eliminates the undesirable feature of high droop controls by automatically compensating for the droop and thereby making it unnecessary for the pilot to reset the desired speed each time a large power change is made.

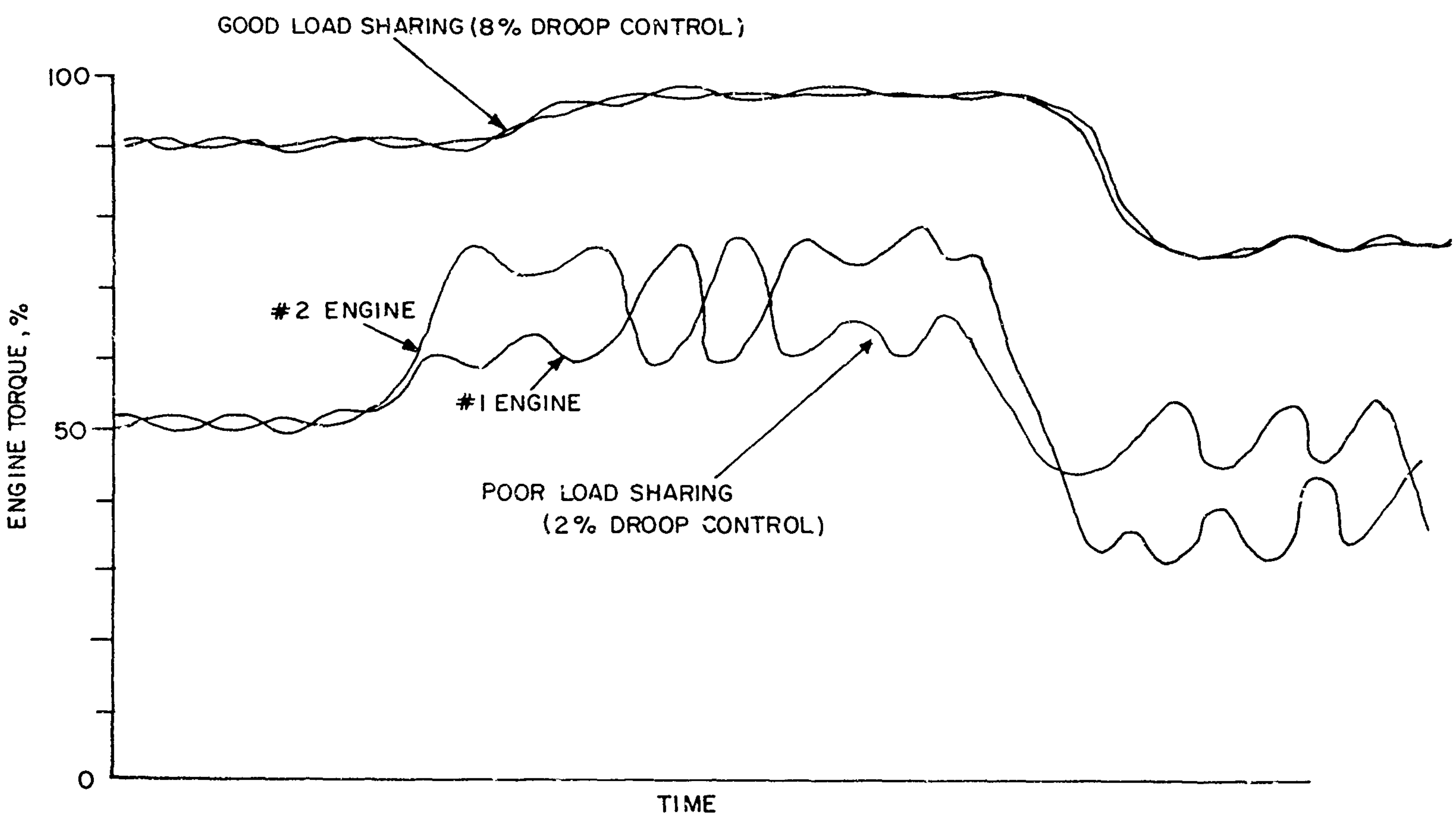


Figure 40. Engine Torque vs Time.

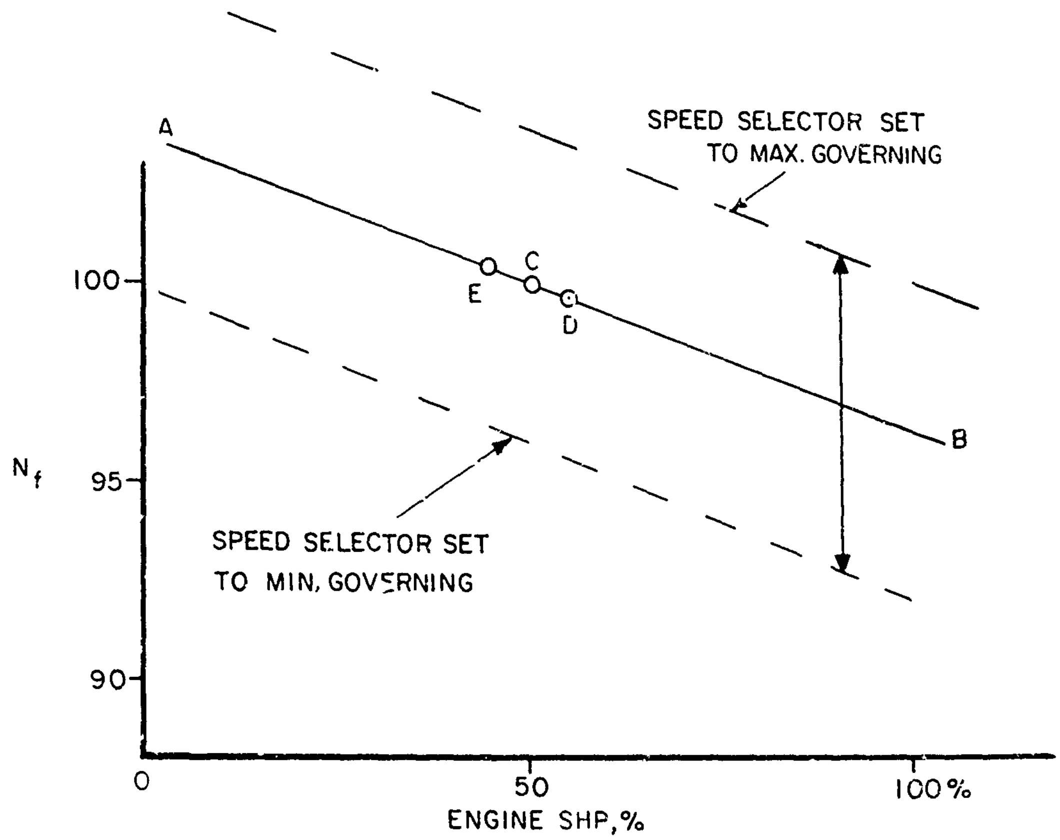


Figure 41. Typical N_f Governor Droop Line.

Problem 12. S-64 (German Version)/JFTD12A: Feedback Torsional Oscillation of N_F Cable Mode

Manifestation of Problem

The problem occurred during the S-64 flying crane prototype development program in 1962. The S-64 prototype was powered by two Pratt and Whitney JFTD12A free-turbine engines.

The first indication of the problem was an in-flight flameout of one engine. As a result of the flameout, a precautionary landing was made. During the landing the container cargo load which was suspended from four hard points under the fuselage struck a large rock. The resulting impact caused a structural failure of the tail cone; however, no other aircraft damage or crew injury resulted.

Investigation of Problem

Inspection of the engine and helicopter following the incident did not reveal the cause of the engine flameout; therefore, it was decided to conduct a ground test program which would attempt to duplicate the problem.

The two engines were removed from the aircraft and installed on the S-64 Propulsion System Test Bed (PSTB). The PSTB is a test facility which duplicates the helicopter propulsion system, engine main gearbox, intermediate gearbox, tail gearbox, drive shafting, main rotor, tail rotor, and flight control system mounted on a steel framework.

Prior to starting the test, the engines were instrumented with the following parameters which were recorded on an oscillograph:

- Gas Generator Speed (N_1)
- Free Turbine Speed (N_2)
- Compressor Discharge Pressure (P_3)
- Turbine Inlet Temp. (T_S)
- Fuel Flow (W_f)
- Fuel Control Discharge Fuel Pressure
- Fuel Supply Pressure
- Engine Torque
- Main Rotor Torque
- Tail Rotor Torque
- Main Rotor Speed (N_R)
- Engine and Flight Controls Positions
- Vibration Transducers

The initial test run was planned to duplicate, as close as possible, the flight during which the flameout incident occurred. At the point in this test when the N_2 controls were advanced to maximum (to produce an N_2/N_R speed of approximately 106-107%), a flameout of the number 1 (L.H.) engine occurred (same engine that flamed out on the aircraft).

Examination of the oscillograph records taken prior to and during the flameout revealed that just prior to the fameout, an oscillation at 11 Hz was observed primarily in fuel pressure. The 11 Hz was evident in most of the other engine parameters but at a very low amplitude. The 11-Hz fuel pressure oscillation increased in amplitude, as shown in Figure 42, and then reduced to zero at the moment of flameout. Based on the sudden decrease of fuel pressure to zero, it was suspected that the overspeed had been actuated. The overspeed system on the JFTD12A engine is actuated when the N_2 governor senses 115% speed. When actuated, the fuel shutoff valve within the fuel control closes and latches in the closed position. To unlatch the shutoff valve, the N_1 control must be moved to the stopcock (shutoff) position. Then the engine can be restarted. A check was made of the number 1 engine and it was verified that the overspeed shutoff had actuated and latched in the closed position.

Since the engine was operating well below the 115% N_2 limit at the time of flameout, it was speculated that the N_2 governor was being subjected to speed oscillations about its set point of 107% and that one of these oscillations peaked above 115%, thereby actuating the overspeed system.

Special instrumentation was installed to measure speed at the N_2 governor drive. Tests conducted with this special instrumentation installed verified that speed oscillations were occurring at the governor drive input. The frequency of these oscillations was 11 Hz and it was demonstrated repeatedly that each time a speed oscillation peak occurred at or above 115%, the overspeed was actuated and flameout occurred (see Figure 43).

Additional tests were conducted to determine the torsional natural frequency of the N_2 governor flex drive cable (see Figure 44). These tests confirmed a torsional natural frequency of 11 Hz.

Testing with other fuel controls of the same part number revealed that this problem could not be duplicated with all fuel controls. It was found that approximately 60% of the fuel controls were good and 40% were unsatisfactory.

An experiment was designed in which parts from a good and bad control were exchanged in accordance with a matrix plan. Results of these tests revealed that the problem would follow the N_2 governor bearing. This bearing was a combination sleeve bearing and seal of carbon material. Detailed inspection of the good and bad bearings revealed both to be within drawing tolerances, and no measurable difference between the two bearings could be determined. It was concluded that for some undetermined reason the good bearing was apparently providing increased torsional damping as compared to the bad bearing.

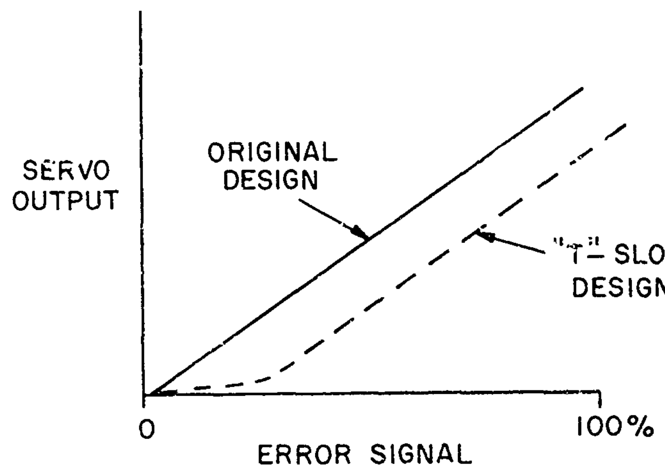
Nature of Problem

Based on the results of the above tests, it was apparent that the engine and fuel control were responsive at 11 Hz and that once excited by any of a number of possible excitations, the fuel control would feed back and excite the torsicnal natural frequency of the N_2 flex drive cable, causing an increasing amplitude torsional oscillation at 11 Hz.

Solution

1. The N_2 governor flex drive cable installation was redesigned to shorten its length and thereby raise its torsional natural frequency.

2. The N_2 governor servo valve was modified to a "T" slot configuration. This modified valve provided a much reduced gain for small error signal inputs but retained high gain for larger error signal inputs, as shown below:



This change made the fuel control unresponsive to the higher frequency (5-15 Hz) inputs but retained fast response for larger step-type inputs.

Subsequent to incorporation of these modifications, no other occurrences of the subject problem have been experienced.

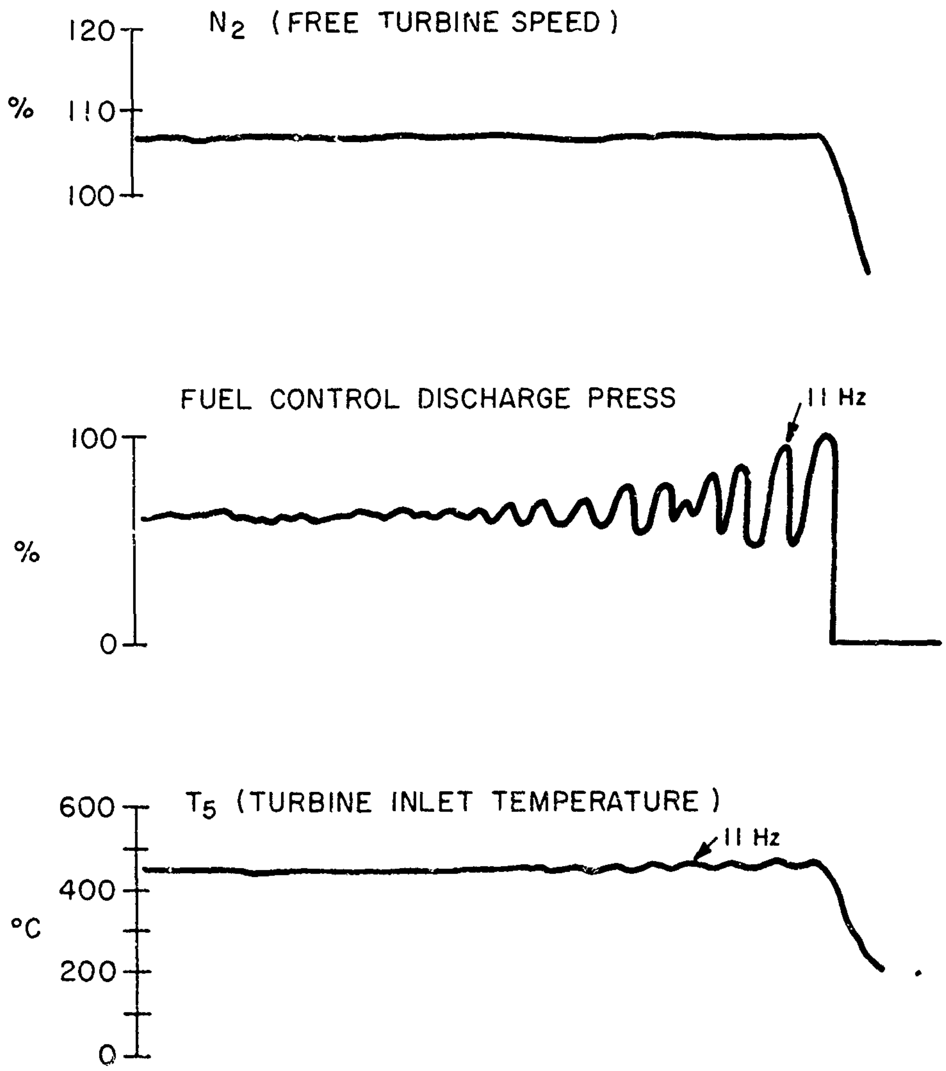


Figure 42. Simulation of In-Flight Shutdown on PSTB.

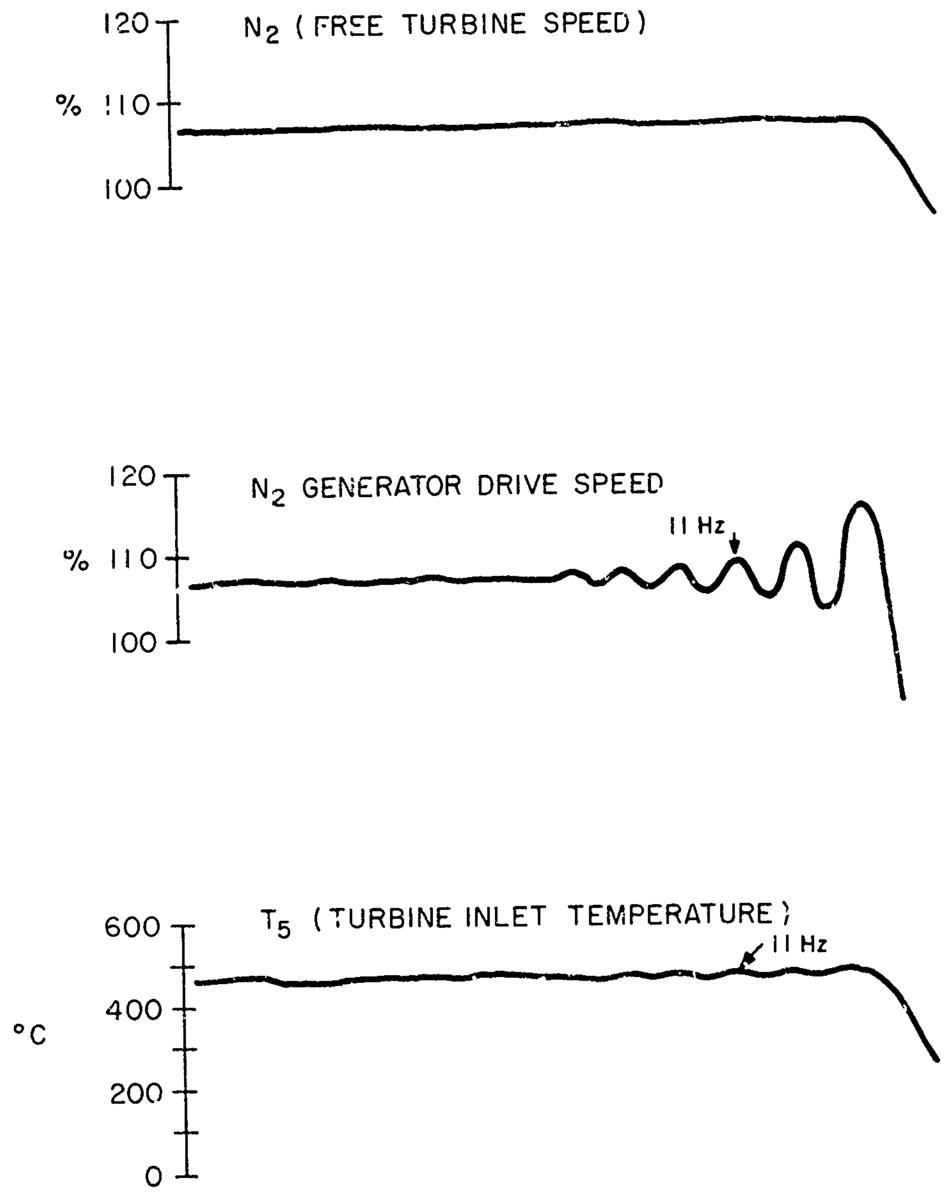


Figure 43. Measurement of N_2 Governor Drive Speed on PSTB.

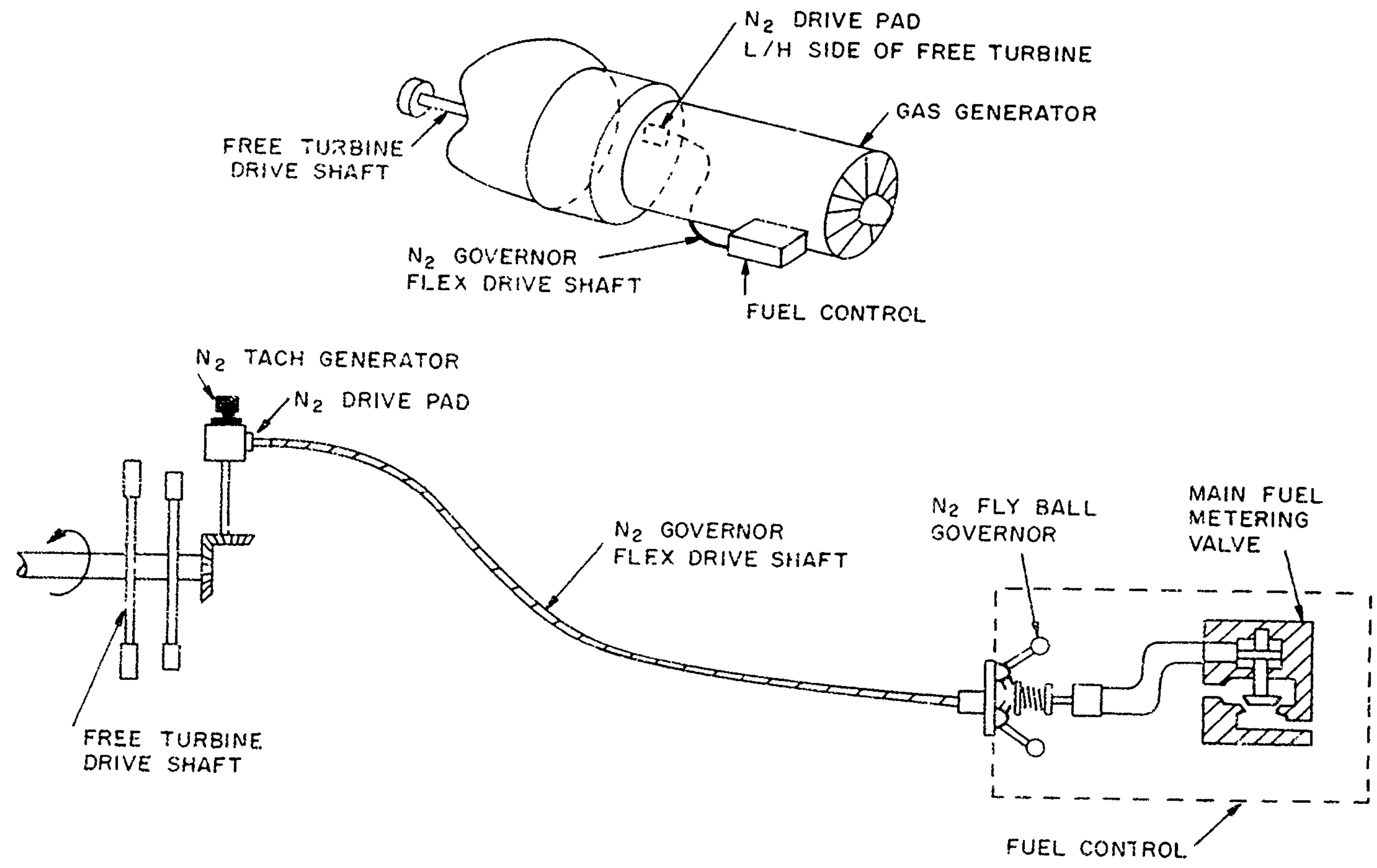


Figure 44. Schematic of N_2 Governor Drive System.

Problem 13. CH-53A/T64: Drive Shaft Supercritical Hysteresis Oscillation

Problem Manifestation

During initial ground running of the CH-53A in August 1964, prior to the start of tiedown endurance testing, the right (number 2) main transmission input drive shaft failed. The failure occurred after approximately 2 minutes of operation at a topping power of approximately 2600 horsepower per engine. At the time of the failure, the tiedown aircraft had accumulated 1 hour 19 minutes of rotor-running time at powers progressively increasing to the topping power. The shaft failure caused limited damage to the airframe, main gearbox, and the number 2 nose gearbox and engine.

Background

The original design of the main transmission input shaft was of two sections with a viscously damped bearing at the intermediate support (Figure 45). Since the CH-53A engine-nose gearbox combination is mounted on vibration isolators, it was necessary to provide for angular and axial freedom in the shaft. This was accomplished by gear couplings (manufactured by Zurn Industries) on each end of the nose gearbox section of the shaft and a single laminated disk coupling at the main transmission input. The gear couplings have sliding splines which allow adjustment for both shaft length and angular misalignment.

The input drive shaft system operates above the first damped rigid body mode and below the first bending mode. In the preliminary design phase a one-piece input drive shaft was considered. While the one-piece shaft would eliminate the rigid body whirl mode, an 8.5-inch-diameter tube would be required to stay sufficiently below the first bending mode. At that time, tubes of this diameter were not available to the degree of straightness required. Multiple section drive shaft systems running above the damped rigid body mode had been successfully used by Sikorsky Aircraft in the tail rotor drive systems of the CH-3C, CH-53A, and CH-54A helicopters.

Prior to the shaft failure incident, the input drive shaft sytems had accumulated in excess of 50 hours of endurance test time on the CH-53A power transmission test bed. The testing included operation at various engine powers up to the maximum obtainable. During these tests, the vibration level of the input drive shaft center bearing support was monitored and no resonance was detected.

Investigation

Immediately following this incident all the parts of the number 1 and 2 input drive shaft installations were subjected to an engineering inspection including an analysis of the failed parts by the Sikorsky Aircraft Materials and Process Laboratory. This inspection revealed:

1. The number 2 shaft support strut end fittings had failed in fatigue. In addition, the end fitting of the inboard strut on the number 1 shaft support had a fatigue crack. This crack was located in the same area as the fatigue failures in the number 2 shaft support strut assembly.

2. Metallurgical examination revealed that the support strut end fittings were in conformance with the drawing requirements.

3. Examination of the remaining components of the number 2 shaft installation indicated that other damage was secondary to the support strut failure.

Using additional instrumentation, the vibration level of the input drive shaft center bearing support was investigated on the power transmission test bed. No excessive vibration could be detected. Next, the tiedown aircraft was placed into operation with gear couplings, redesigned center bearing support strut fittings, and considerable center bearing support instrumentation. Testing showed a 30 cps (approximately) vibratory stress in the center support structure. The magnitude of this stress increased with increasing drive shaft torque but remained nearly constant with shaft speed.

An effort was made to increase the damping capability of the center bearing support by increasing the rigidity of the support strut assembly on the tiedown aircraft. This resulted in decreasing the sensitivity of the vibratory stress to the shaft torque, but the changes did not result in a completely stable system.

Since the 30 cps vibration was at the natural frequency of the first (rigid body) mode of the two-piece shaft system, it was concluded that vibration was being excited by the system hysteresis resulting from the internal friction of the gear coupling. A discussion of shaft vibration caused by hysteresis, when the shaft is rotating faster than the first critical speed, is given in Reference 1. From this reference it may be seen that the effect of hysteresis is to excite the shaft vibration at the mode natural freuqency even though the shaft is rotating faster than the mode critical speed.

To verify these conclusions, the tiedown aircraft gear couplings were replaced with diaphragm couplings (manufactured by the Bendix Corporation) available from an in-house development program. The couplings accommodate the length changes and shaft angular misalignment through several series-connected diaphragms. Since the diaphragms have negligible internal friction, the drive system would approach zero hysteresis. The tiedown aircraft successfully completed 20 hours of endurance testing using the diaphragm couplings. After the completion of the endurance test, these couplings were installed on the first flight aircraft (BuNo. 151614).

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1. Timoshenko, S., Vibration Problems in Engineering, 3rd edition, D. Van Nostrand Co., Inc., Princeton, N.J., 1955, p. 227.

As a final confirmation of the above conclusions, the gear couplings from the power transmission test bed were installed on the tiedown aircraft. Considerable testing failed to produce any 30 cps vibration of the center bearing support. It was concluded that the particular gear couplings originally installed on the power transmission test bed had an internal sliding friction level low enough to prevent exciting the rigid body mode. Inspection of the mating surfaces of these couplings tended to indicate a freer fit than the couplings originally installed on the tiedown aircraft. Probably some of this could be attributed to the break-in by over 50 hours of endurance test time on the power transmission test bed.

Solution

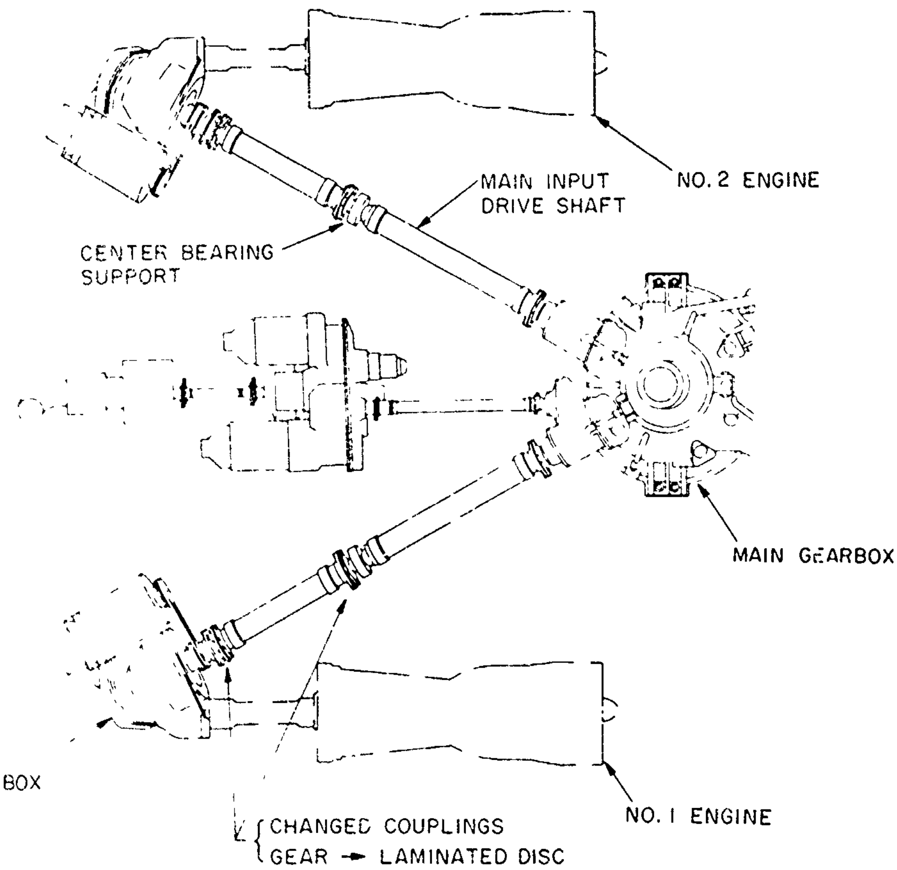
From the testing accomplished, accurate information was obtained as to the actual dimensional changes associated with operation of the input drive shaft system. The magnitude of these changes was such that they could be accommodated by an all-laminated Thomas-type disk coupling configuration. It should be noted that a double disk laminated coupling is used at the nose gearbox output end of the shaft, a single laminated disk coupling is used adjacent to the viscously damped bearing, and the single laminated disk coupling is retained at the main gearbox input end of the shaft. These laminated disk couplings accommodate the length and angular changes by plate flexure; therefore, it would be expected that these couplings would demonstrate the desirable low internal friction properties.

In the initial redesign, a ball joint was used between the double laminated disks. The ball joint was used to place the critical speed of the rigid body whirl mode of the new input drive shaft at 185 percent of normal operation speed. This configuration successfully passed the critical speed test and accumulated 30 hours of tiedown endurance testing.

The input drive shaft configuration was further optimized by shortening the double disk interconnecting shaft and eliminating the ball joint. In this configuration the rigid body critical speed, for the above described mode, remained at 185 percent of the normal operating speed.

This shortened double laminated disk configuration successfully accumulated 50 hours of the tiedown endurance testing and was installed as the production configuration

During the testing of the various input drive shaft coupling configurations on the tiedown aircraft, the vibratory stress in the center bearing support strut attachment fittings was monitored. The test results conclusively demonstrated that all of the negligible internal friction couplings (diaphragm, double laminated disk - ball joint, shortened double laminated disk) eliminated the 30 cps vibratory stress. The test results are summarized in Table 4.



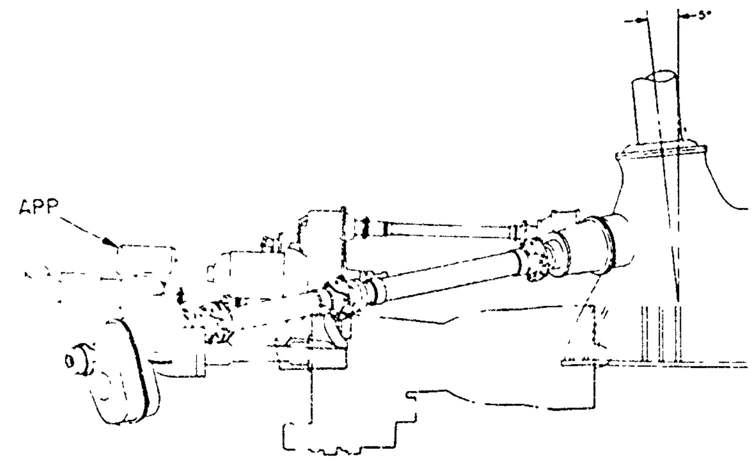


Figure 45. CH-53A Transmission System.

TABLE 4. TEST RESULTS, INVESTIGATION OF INPUT SHAFT VIBRATION ON CH-53A TIEDOWN AIRCRAFT

COUPLING CONFIGURATION	FREQUENCY	SHAFT WHIRL DISPLACEMENT	VIBRATORY STRESS IN CENTER BEARING SUPPORT STRUT ATTACHMENT FITTINGS
Gear	1 X Drive Shaft Speed	Less than ± 3 mils	Less than ± 1000 psi
	30 - 36 cps	± 110 to ± 150 mils	± 18,000 to ± 20,000 psi
Diaphragm or Double Laminated Disk Ball Joint or Shortened Double Laminated Disk	1 X Drive Shaft Speed	Less than ± 3 mils	Less than ± 1000 psi
	30 - 36 cps	None	None

Problem 14. XH-59A/PT6T-6: Engine Hunting Oscillation

Problem Manifestation

During functional and operational tests of the XH-59A/PT6T-6 propulsion system in February 1973, a hunting oscillation was observed at low frequency and low rotor speeds.

Nature of Problem

The XH-59A propulsion system consists basically of two contrarotating three-bladed rotors, gearbox, and PT6T-6 engine. This system was undergoing functional and operational tests on the PSTB facility. The test stand configuration is shown in the drawing of Figure 46. In February 1973 an oscillation was discovered which appeared to be a classical hunting stability problem. The oscillations were nondivergent and occurred in the frequency range of 1/3 to 1/2 Hz, at approximately 70% N_R. Apparently the usage of too high a governor gain led to the oscillation.

The problem was caused by the unusual capability of the ABC aircraft. This aircraft has a large range in forward speed, zero to 280 knots. At high aircraft speeds, the rotor speed must be reduced to keep the blade tips from going supersonic. This causes an extremely large rotor speed range for governing, i.e., approximately 55 to 105% N_R. This range compares with the 95 to 105% N_R range for conventional helicopters.

The engine control approach was to alter the power turbine governers to provide the required speed range and alter the Pg accumulator volume to adjust governor time constant. To meet the required low speed governing condition, the N_f governors were altered to provide a droop of 5% at maximum speed. This is almost twice the normal PT6T-3 engine gain and about 2½ times the nominal T400 engine gain. This high level was selected for two reasons: to allow governing down to 55% N_f at moderate power and at this low speed to provide sufficient gain to govern effectively.

Problem Investigation

Subsequent to the initial oscillation, a variety of modifications were made to the engine control. Changes were made both in governor gain and time constant, and different governor models were tried. Considerable improvement was made although it was not completely successful due to the adverse effect of the torque equalizer which was still connected, and also due to the large rotor speed range required. The torque equalizer was a control unit used to ensure equal load sharing between the engines.

Final Solution

The solution developed for the aircraft and the PSTB was to reduce the governor gain to 9% droop and remove the torque equalizer from the control loop. By special calibration it was possible to bench calibrate governors and controls as matched pairs resulting in small torque splits. This condition eliminated the need for a torque equalizer; however, a differential "beeper" was provided so that the pilot could null out any of the small splits that may have been objectionable.

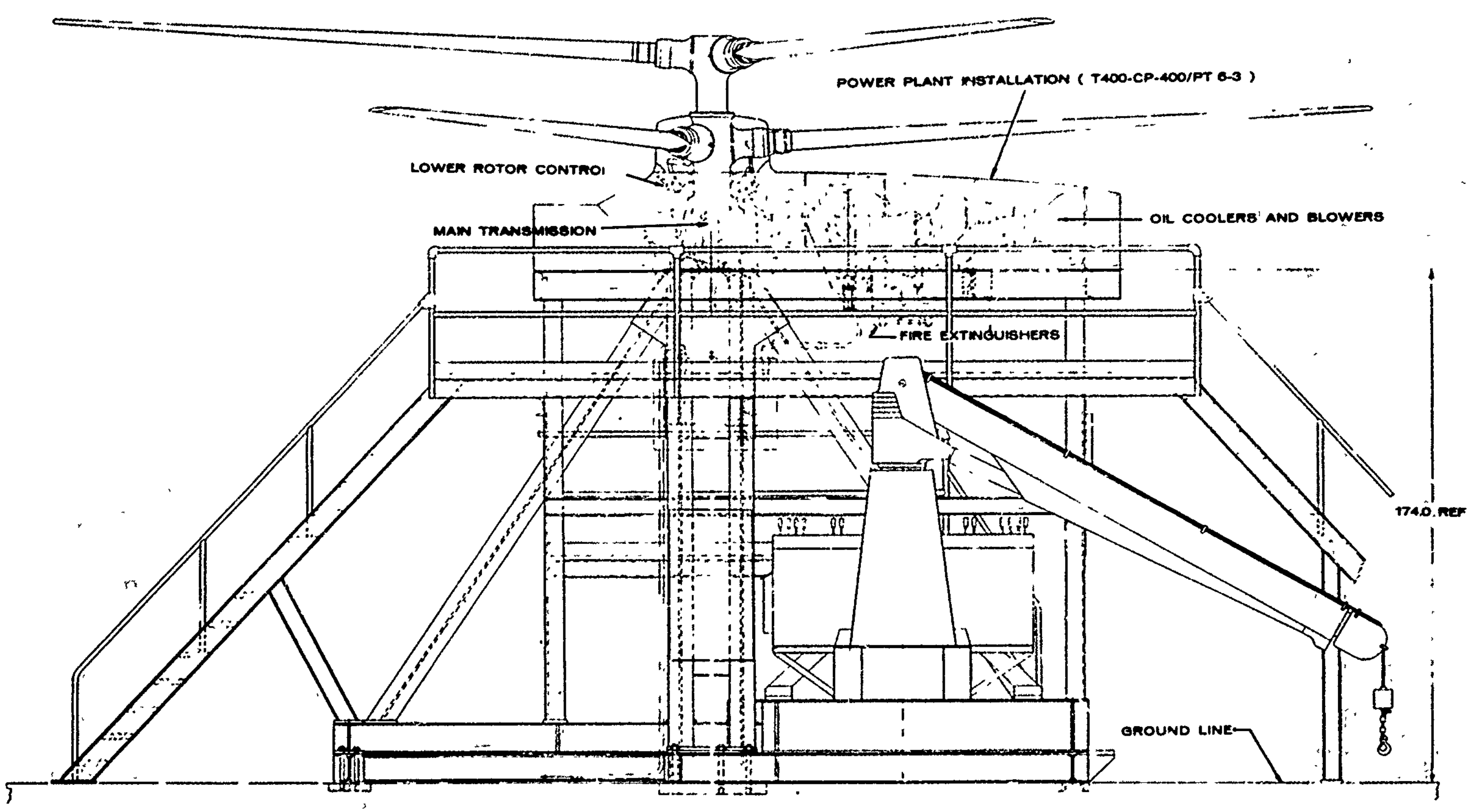


Figure 46. XH-59A Propulsion System Test Stand.

Problem 15. Twin-Engine (1500 shp) Helicopter: Minor Engine Hunting in Accessory Drive Operation Only

Problem Manifestation

During ground operations, in accessory drive, an estimated 0.5-0.7 Hz oscillation exists on the number 1 engine of a twin-engine 1500 shp helicopter. Accessory drive is the condition whereby the number 1 engine is driving through the main gearbox to run electrical/hydraulic accessories <u>only</u>, not driving the main rotor - this is a very low power requirement. When the number 1 engine is driving the rotor head, no oscillation exists. This phenomenon is noticeable in minor cycling fluctuations of the engine instruments and in a slight audible sense; it is more prominent in some aircraft than in others.

Solution

After several attempts at resolution by the engine manufacturer over a 3-year period, the customer has decided to accept the situation as "standard-to-type", and has dropped further pursuits for solution.

The engine manufacturer's position is that no specific design criteria are required to assure that there is no oscillation at such low power conditions since it is not detrimental.

Problem 16. CH-53E/T64-GE-415: Oscillation of Rotor/Drive System 3rd Torsional Mode

Manifestation of Problem

During developmental flight testing of the YCH-53E Aircraft S/N 159122 in 1975, a limited amplitude oscillation of the aircraft at 3.6 per main rotor (10.5 Hz) was encountered. A recent configuration change had stiffened the main rotor blades by the addition of graphite strips on the trailing edges to improve blade stress levels. The self-excited vibration occurred in a very limited range of operating conditions (forward speed 90 to 120 knots, partial power descent, 500-1500 rpm, and gross weight less than 46,000 lbs, neutral to forward c.g., 99-100% N_R). Cockpit response of approximately $\pm$ 0.25 g was measured. Time to double amplitude for the vibration was 10-12 seconds. The stresses and loads appearing throughout the aircraft at that frequency were low and were not significant. The response was usually obtained by lowering collective pitch to enter a descent, and the response could readily be stopped by increasing or decreasing collective pitch.

The Self-Excited Mode

Study of test and analytical data showed that the self-excited mode was a torsional mode comprised of blade edgewise bending, with all seven blades acting collectively, and drive system torsional motion. Node lines were on the blades (at 3/4 radius), in the main rotor shaft, and in the tail rotor shaft (Figure 47). The blade motion combined rigid body lag and first bending mode, summing to give little motion across the lag damper (solid line of Figure 48). A ground test was conducted in which the flight controls were oscillated at varying frequencies and the rotor response was measured. The test confirmed the existence of a mode at 3.6P. Still, sustained oscillations of the type experienced would not have been present if this mode were not being excited by coupling to a source of energy.

Energy Source

One source of this energy could have been in the fuel control system, as in Problem 17. However, analyses performed by General Electric seemed to rule this out. Figure 49 shows a frequency response plot of the open loop rotor/drive train/fuel control system as calculated by General Electric. The mathematical model used for this analysis was essentially the same as that described in Problem 17 (nine lumped inertias plus the fuel control system), except that the first blade flexural edgewise bending mode was added as a blade bending degree of freedom (Figure 50). As seen in Figure 50 the gain margin is very large (30dB) in the region of 3.6P (66.7 rad/sec), indicating a large margin of stability.

Another possible energy source was aerodynamic. This was not ruled out by the stable GE analysis, since that model had no blade flapping or twisting degrees of freedom. Dr. Raymond G. Carlson of Sikorsky postulated that the self-excited vibration was due to a coupling between the 3.6P edgewise collective rotor mode and a 2.6P (in the rotating system) flatwise forward whirl rotor mode (Reference 2). Forward speed and torsional windup provided the coupling between the modes.

Possible Mechanism

The following defines this mechanism in more detail.

The coupling of a whirl mode (cyclic mode) with a collective mode at a frequency which is a noninteger multiple of rotor speed requires two conditions:

1. A 1P mechanism which can add 1P to 2.6P to give 3.6P and conversely subtract 1P from 3.6P to give 2.6P.

2. A coupling which will yield an interaction which is not a function of azimuth angle.

This coupling is provided by airspeed (advance ratio) and torsional wind-up.

The loop which describes the interaction is shown in Figure 51. The 2.6P flatwise mode provides a loading of the 3.6P collective mode. A forward whirl mode has the characteristic that a blade reaches a maximum response later than the blade which follows it. Thus, a wave of peak response is produced which travels around the azimuth at 2.6P in the rotating axis (or 3.6P in fixed axis). This mode produces a torsional windup or pitch change at 2.6P. The pitch change is acted upon at forward speed by a 1P dynamic pressure variation which is of opposite phasing. That is, the dynamic pressure on a blade reaches a maximum value (at azimuth 90^{0}) earlier than on the blade which follows it. The combination of pitch variation and dynamic pressure produces a 3.6P loading which is independent of azimuth angle and is in phase for all blades. This loading produces a response of the 3.6P collective edgewise blade/drive system mode.

The 3.6P collective mode produces a loading of the 2.6P flatwise mode. The argument is similar to the above. The 3.6P response causes a 3.6P pitch variation, which is in phase on all blades and acts on the dynamic pressure variation at 1P. This combination produces a 2.6P loading

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2. Hilzinger, K., Twomey, W., and Molnar, G., Model CH-53E Vibration and Flutter Analysis and Test Report, Engineering Report SER-13178, Sikorsky Aircraft Division, United Technologies Corp., April 1976.

which has the proper phasing for the excitation of the 2.6P forward whirl mode. Thus, the loop is completed, with two modes exciting each other.

The aircraft responds at 3.6P vertically for two reasons. The 3.6P collective mode has significant flatwise blade deflection causing vertical excitation of the aircraft. The 2.6P whirl mode produces a 3.6P pitch and roll response.

Analytical Investigation

Calculations of the main rotor blade modes for blade set number 4, which were the blades on the aircraft when 3.6P oscillation was encountered, gave a first flatwise bending mode at 2.7P and a first edgewise bending mode at 4.05P as shown in Figure 52. The seven blades acting collectively (in phase) edgewise combine with the torsional dynamics of the drive train (engines, shafts, transmission) to give a combined rotor drive train mode at 3.76P analytically. This is shown in Figure 53. This figure shows the impedance of the drive train at the rotor hub for the range of frequencies up to 6P (shown in solid) and the natural frequency of the edgewise rotor mode acting against varying hub impedance values (shown as a dashed line). The intersection of the two curves occurs at the natural frequency of the combined rotor/drive train mode.

Test data indicate that the edgewise rotor/drive train mode is at 3.6P and the flatwise mode is in the area of 2.6P, somewhat lower than predicted. These frequencies were measured in a ground shake test of the aircraft with the rotor turning. The edgewise resonance at 3.6P was clearly defined, but the flatwise resonance at or near 2.6P was not well defined.

The 3.6P edgewise mode has very little angular motion across the lag hinge. Therefore, the lag damper is ineffective in providing damping for this mode. The calculated mode shape is shown by the solid line in Figure 48. The hub arm and the root end of the blade move together. The 3.6P mode also has substantial flatwise deflection as shown in Figure 54, which was obtained from the coupled mode analysis. The 2.6P mode, Figure 55, is primarily flatwise. The edgewise motion is relatively small. The curves show modes at a collective pitch of 11 degrees. The modes show less coupling between flatwise and edgewise motion as pitch is decreased.

Solution

The oscillation was eliminated by softening the edgewise stiffness of the blades. This decreased the natural frequency of the 3.6P mode and increased the motion across the lag damper in the mode. Figure 48 shows the calculated results. This resulted in a 50% increase in modal damping from the lag damper. The analytical natural frequency went from 3.76P to 3.47P. The test frequency drop, as found from ground tests, was from 3.6P to 3.45P.

The blade edgewise softening was accomplished by returning the blades to their original configuration without the added graphite strips. The resulting higher blade stress levels were considered acceptable for this prototype aircraft since the blades were to be redesigned anyway for the production aircraft.

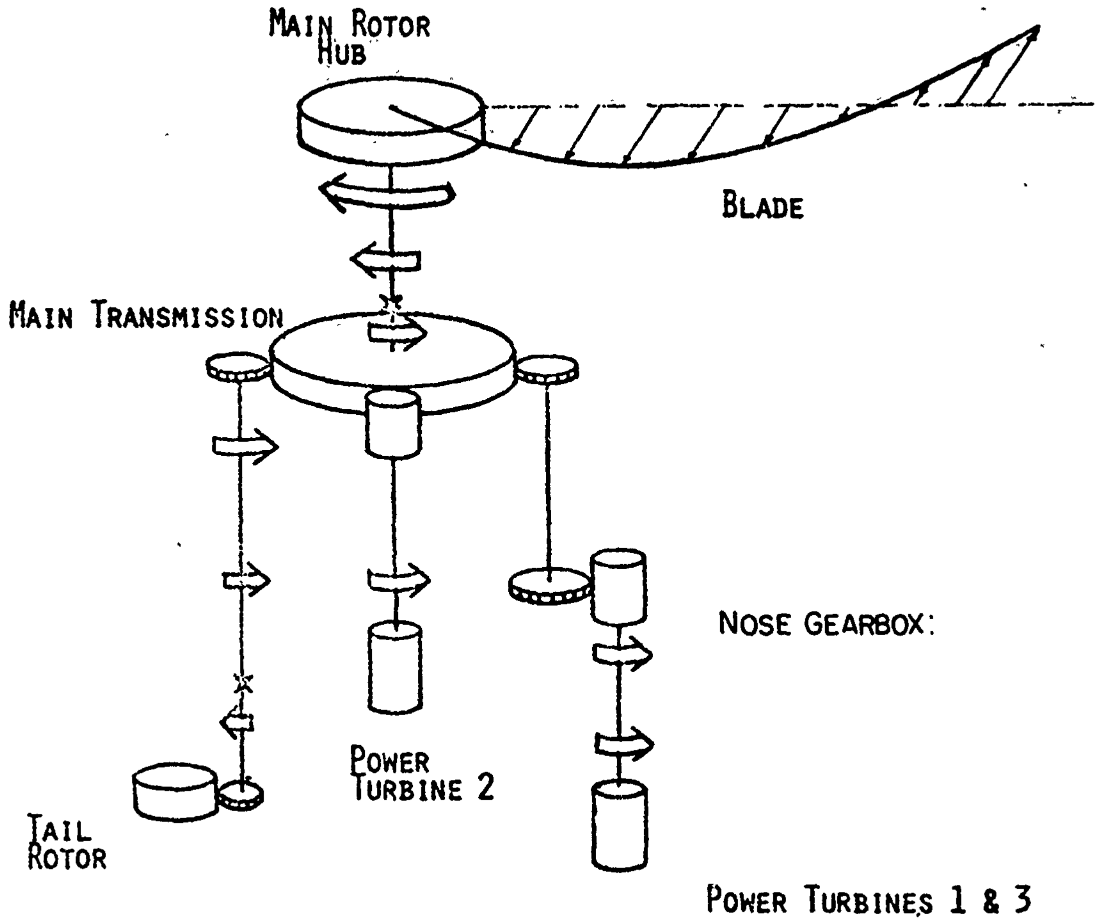

Figure 47. Self-Excited Mode: 3rd Torsional Mode of Rotor/Drive System.

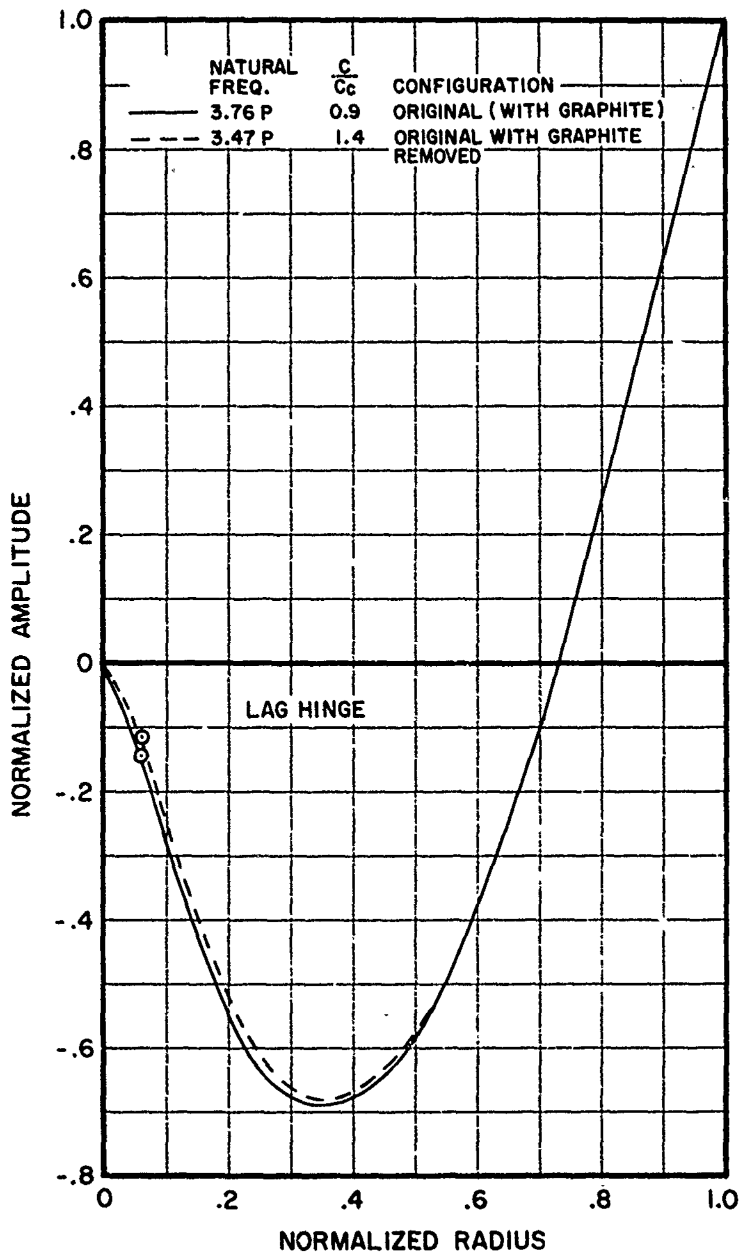

Figure 48. Self-Excited Mode: Detail of Blade Motion.

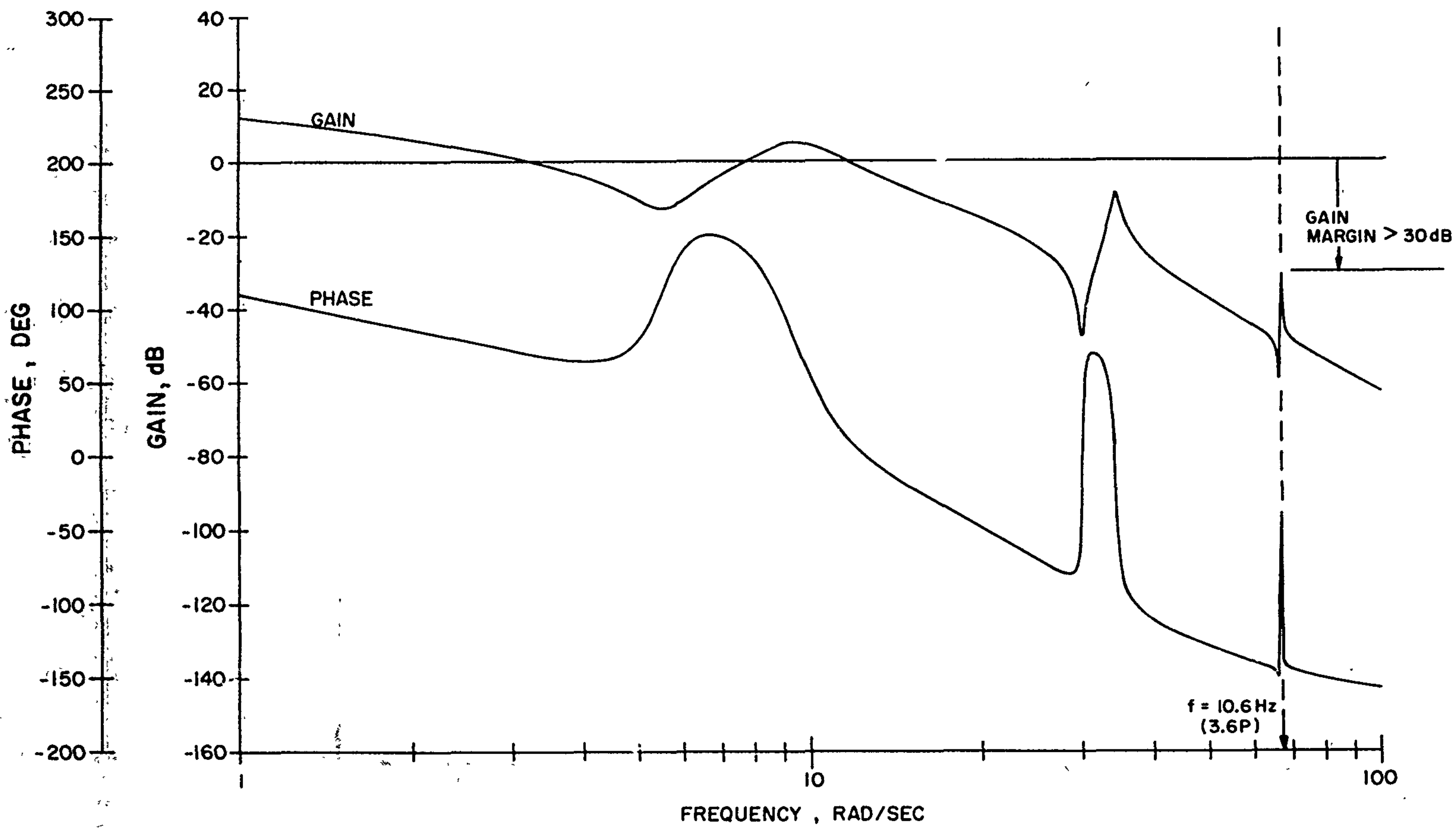

Figure 49. Bode Plot of Rotor/Drive Train/Fuel Control System.

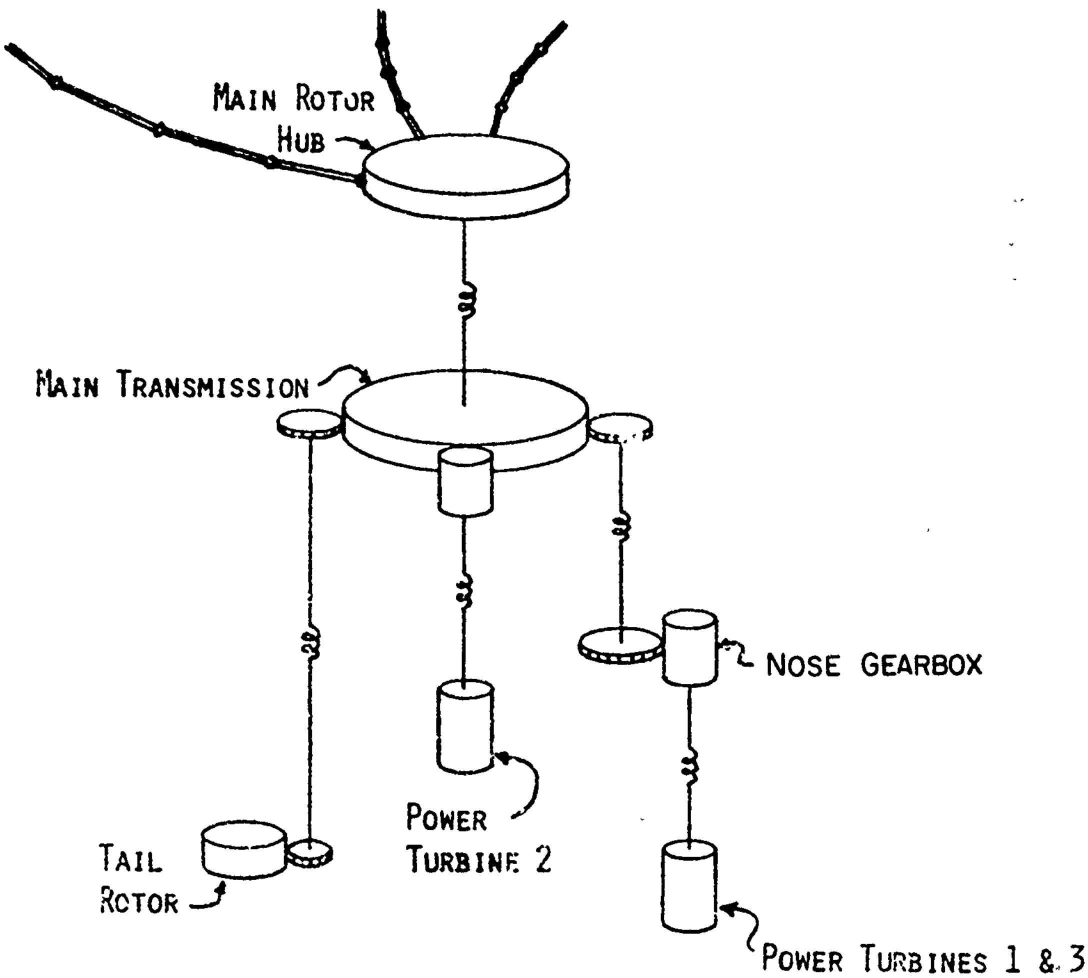

Figure 50. Coupled Rotor/Drive System Analytic Model.

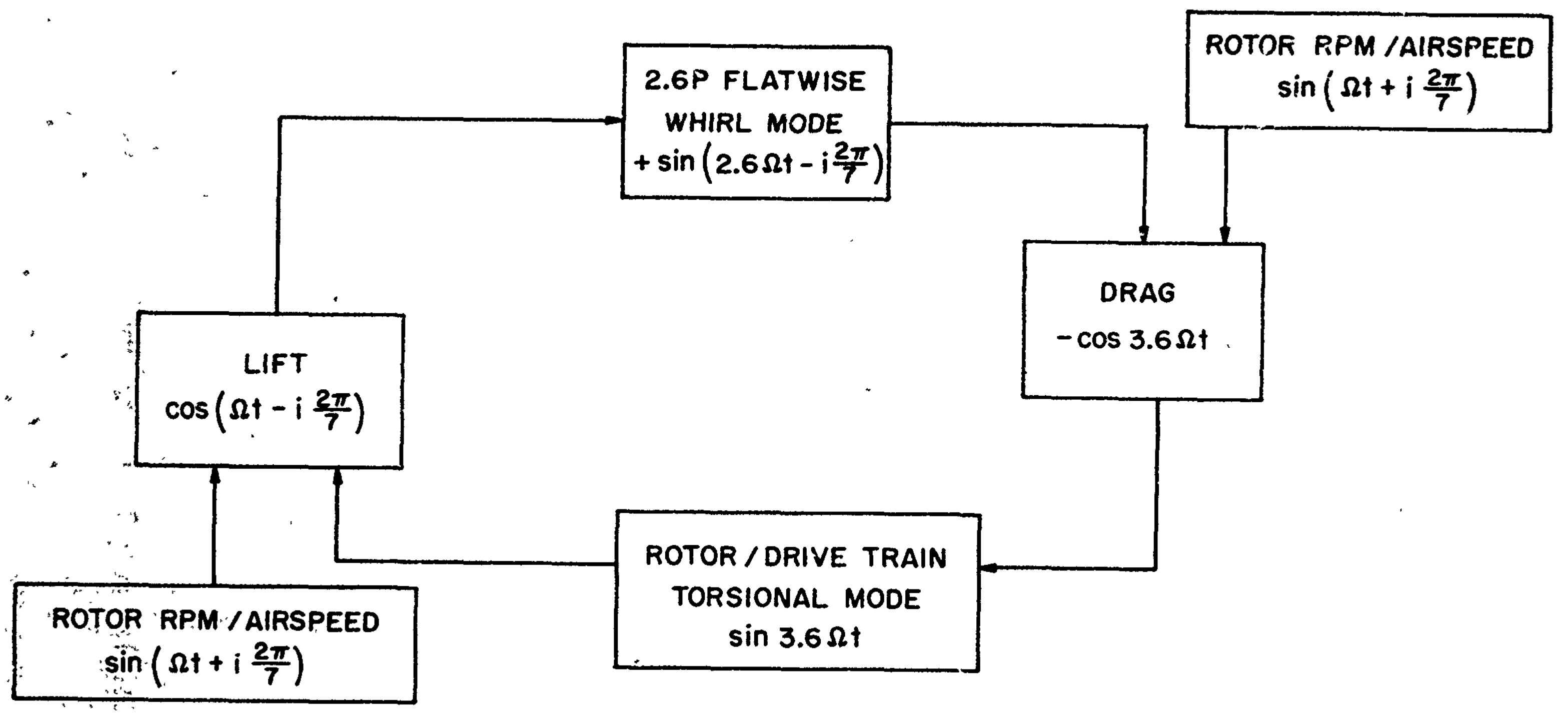

Figure 51. Block Diagram of 3.6P Self-Excitation Mechanism.

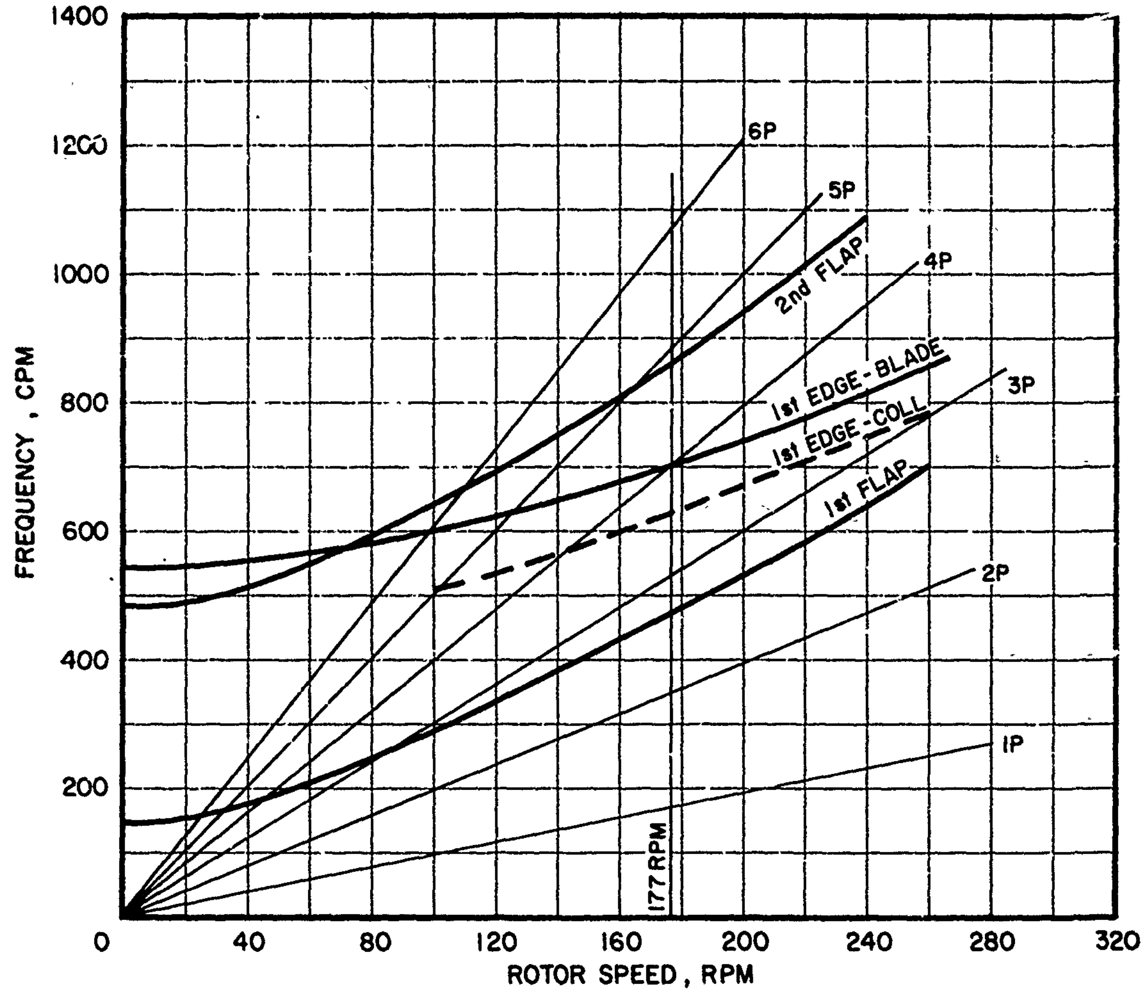

Figure 52. Blade Set Number 4 Natural Frequencies.

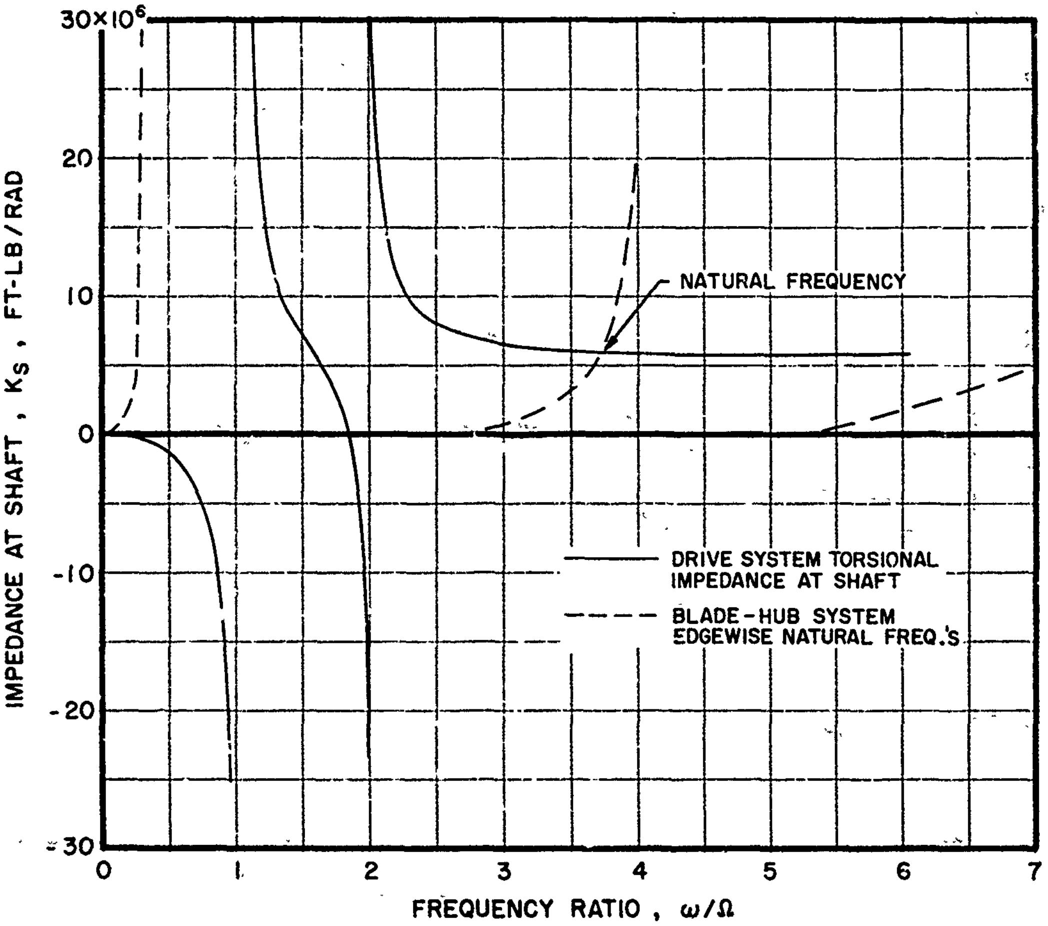

Figure 53. Frequency of Collective Rotor Mode.

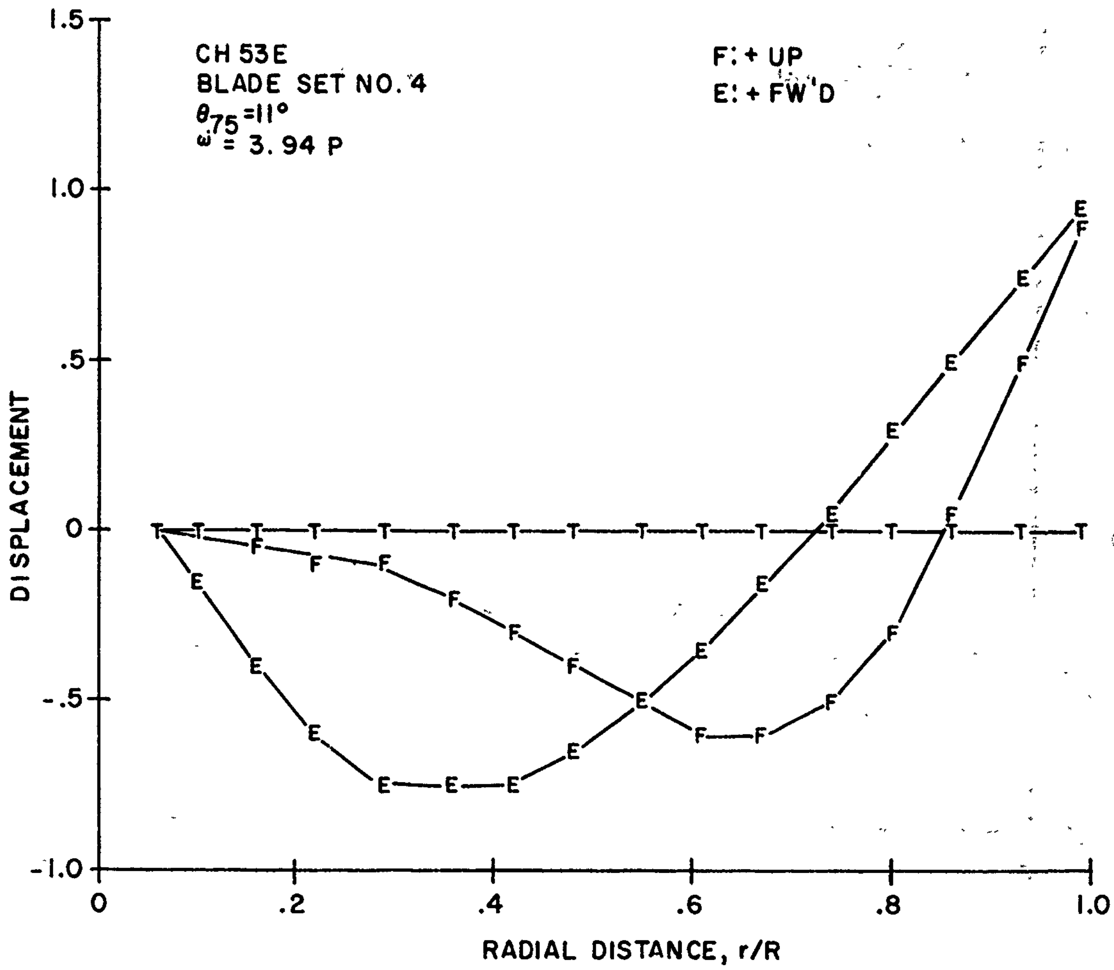

Figure 54. Blade Coupled Flap-Lag Mode Shape (Hub Fixed), Predominantly Edgewise.

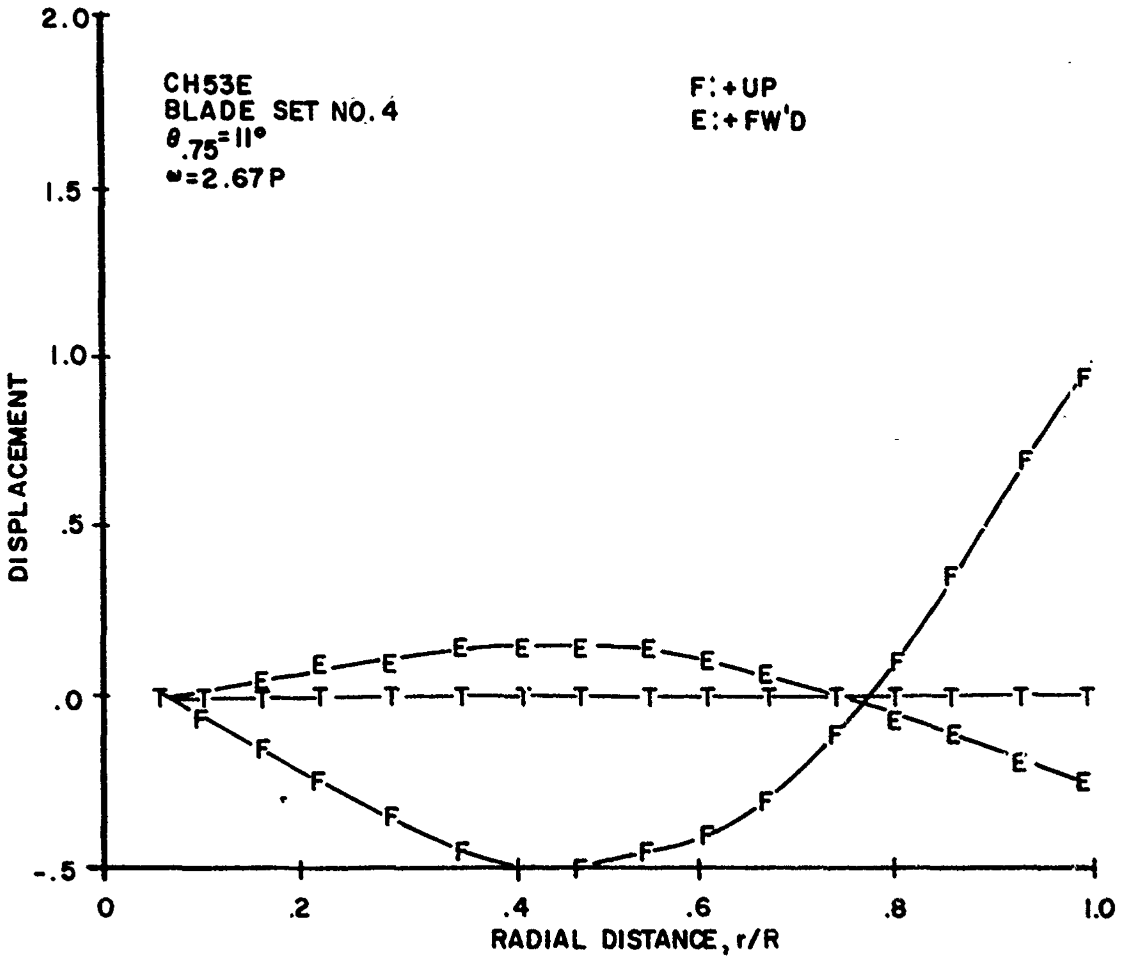

Figure 55. Blade Coupled Flap-Lag Mode Shape (Hub Fixed), Predominantly Flapwise.

PROBLEM 17. CH-53E/T64: Feedback Oscillation of Rotor/Drive System 1st Torsional Mode

Description of Problem

The problem was first encountered during Phase 2 of the CH-53E development program in May 1976 on the 56,000-lb gross weight, forward c.g. configuration. As the aircraft was increasing its airspeed from 150 knots to V_H, a low-frequency (around 2 Hz) oscillation of the aircraft was felt by the pilot. The pilot was able to stop the motion by slowing down the aircraft (pulling back on cyclic stick).

Subsequent analysis and testing showed that the oscillation was a closed loop feedback self-excited oscillation of the collective rotor/drive train/engine/fuel control torsional system. The oscillating mode was the first (nonzero) natural mode of the elasto-mechanical system. In this mode the main rotor blades lag together against the drive train and engines, the only node line occurring on the blades outboard of the lead-lag hinges (Figure 56).

The oscillation was caused by the high gain of the hydromechanical fuel control (rotor speed governor) system feeding energy into the first torsional mode. In this mode the principal dissipation of energy occurs in the blade lag dampers. The effectiveness of these dampers decreases with aircraft forward speed due to the increased presence of superimposed n/rev steady-state motions in the dampers. Eventually a speed is reached at which the damping available is insufficient to dissipate the energy input from the fuel control system. On the CH-53E, this onset of oscillation occurred at different aircraft forward speeds depending on rotor RPM and aircraft gross weight and c.g. position.

Analytical Model

The analytical model used to represent the rotor/drive system was formulated as a 7-mass torsional model (Figure 57). The model had engines 1 and 3 lumped together, and included the rigid blade centrifugal stiffness. The lag damper was represented as a linearized viscous damper. The hydromechanical fuel control/gas generator systems are shown as boxes in Figure 57. A decrease in drive train speed (sensed at the nose gearbox by engines 1 and 3 and at the main gearbox by engine 2) is sensed by the system and results in an increased engine torque being applied (via changes in fuel flow and gas generator speed) to the power turbines.

Stability was determined by calculating the frequency response (Bode) plot of the total rotor/drive train/fuel control open loop system. A positive "gain margin" indicates a stable system, but to allow for manufacturing tolerances a minimum gain margin of 6 dB was considered necessary.

Blade Lag Dampers

Analysis showed that the stability margin was sensitive to blade lag damper values (Figure 58). An extensive test and analytical effort was therefore undertaken to determine just how much damping the lag damper was providing to the critical mode.

The instantaneous force-velocity relationship for the damper is given in Figure 59. In this damper an orifice-type damping force, proportional to the velocity squared, is present up to the "breakout point", at which a preloaded relief valve opens (at a velocity of 0.8 in/sec) and the damping derivative suddenly becomes very much smaller. The effectiveness of such a damper is diminished by the presence of steady-state harmonic (n/rev) lag motions, as seen from bench test data plotted in Figure 60. These motions, which become larger with increasing airspeed (Figure 61), cause the damper relief valve to remain open for increasing portions of the cycle. The resulting loss in damping is shown in Figure 62. Eventually a speed is reached at which the damping available is insufficient to dissipate the energy input from the fuel control system and the overall gain margin goes to zero (Figure 63).

Original analyses had shown adequate stability margins because high values of lag damping available had been used which were inconsistent with the high speed flight condition. More realistic values of lag damping at high forward speed were determined from damper tests and analysis. These brought the analytical stability boundary into good agreement with test, as shown in Figure 64. The analytical modal frequency of 2.26 Hz (14.2 rad/sec) compares with the test values of from 2.0 to 2.2 Hz.

Solution

Once the analytical model was shown to give good correlation with flight test, a parameter sensitivity study was undertaken to find out which parameters most affected stability. The most sensitive parameters were found to be lag damping and those fuel control parameters which reduced fuel control gain.

In practice, sufficiently increasing the lag damping did not prove feasible within the constraints of the hydraulic damper and blade strength.

Decreasing the gain of the fuel control system was first shown to be effective in increasing stability when flight test results of aircraft having fuel control gains on the high and low sides of manufacturing tolerances were compared (Figure 64). Analysis showed that the fuel control gain was most sensitive to:

1) P_3 sensing line time constant and

2) Power turbine (N_F) governor time constant

The fuel control gain could be expeditiously lowered by increasing the diameter of the P_3 sensing line (which increases its time constant). However, this resulted in a deterioration in engine power response to command inputs, which precluded its adoption as a long-term solution. It was used, however, as an interim solution.

Increasing the power turbine (N_F) governor time constant remained the long-term solution. Analysis predicted that increasing the N_F time constant from 0.165 to 0.70 sec. would reduce the fuel control gain at the torsional frequency (Figure 65) enough to produce the required gain margin of 6 dB at the most critical flight condition (Figure 66). Unlike the P_3 sensing line, this parameter lay outside the direct line from pilot collective-compensated power command (collective pitch) to output torque, and thus had little deteriorating effect on engine power response to pilot command. The N_F time constant increase was accomplished physically by decreasing the effective window size of the power turbine governor pilot valve.

Subsequent flight tests confirmed that this solution completely eliminated any self-induced oscillation (modal damping remained above 5% critical throughout the flight envelope) while leaving the engine power response (rotor droop) virtually unchanged.

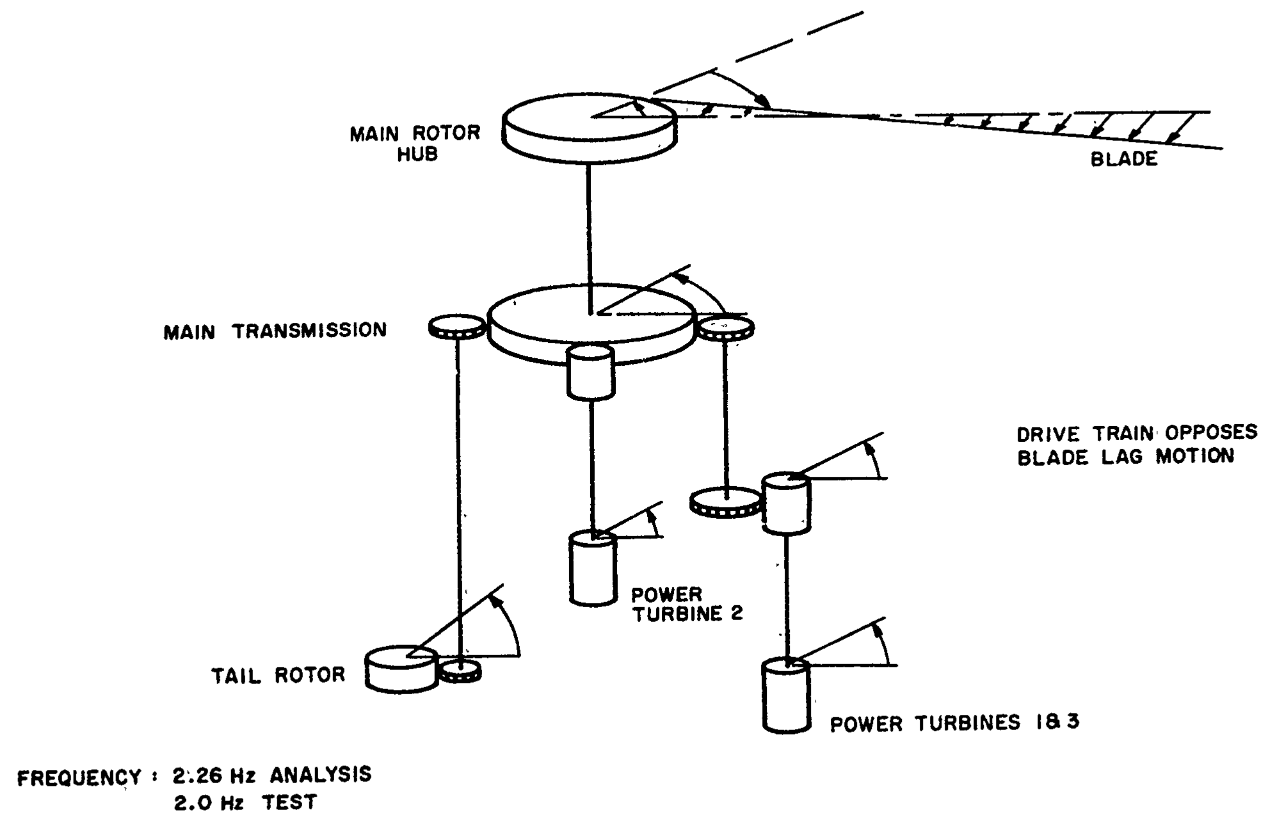

Figure 56. Self-Excited Mode: 1st Torsional Mode of Rotor/ Drive System.

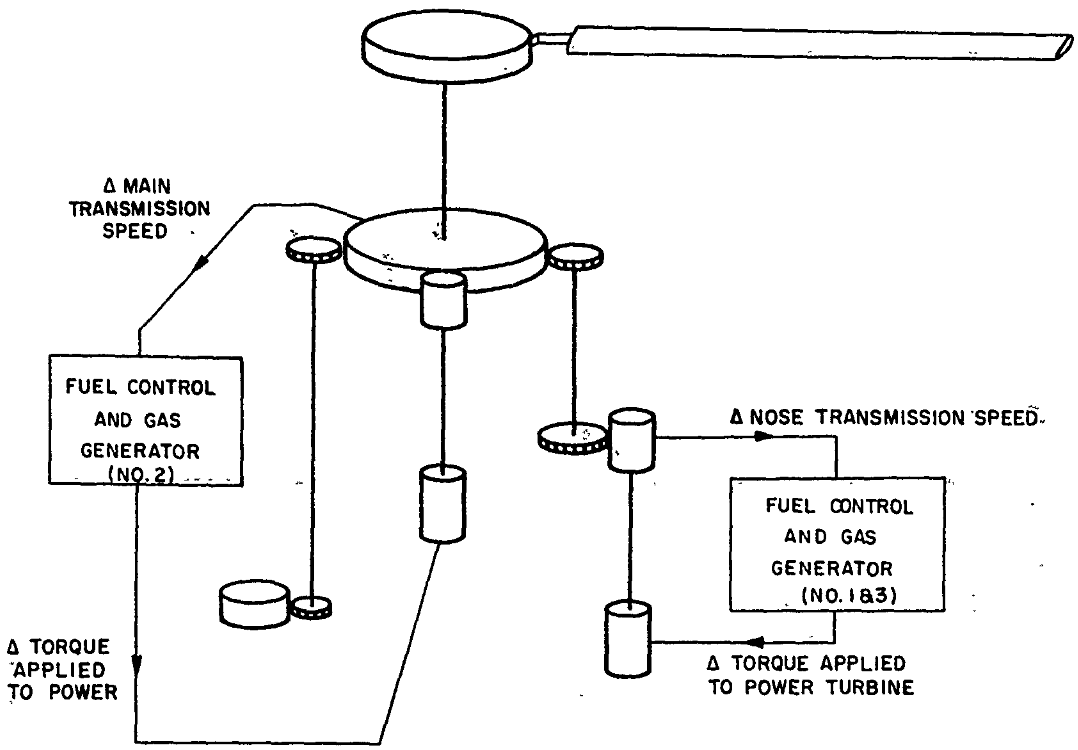

Figure 57. Rotor/Drive System Analytical Model Schematic.

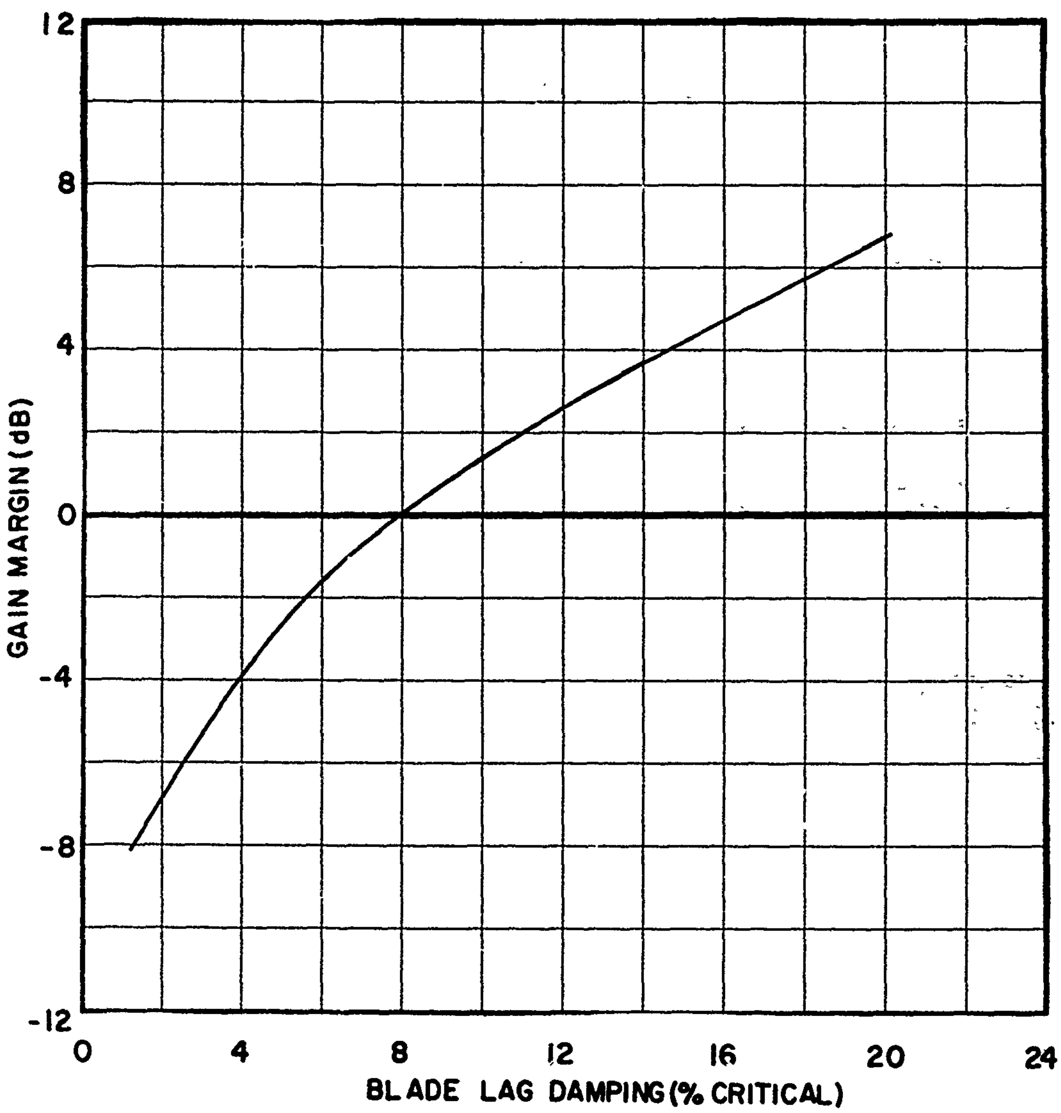

Figure 58. Gain Margin of Torsional Mode vs Blade Lag Damping.

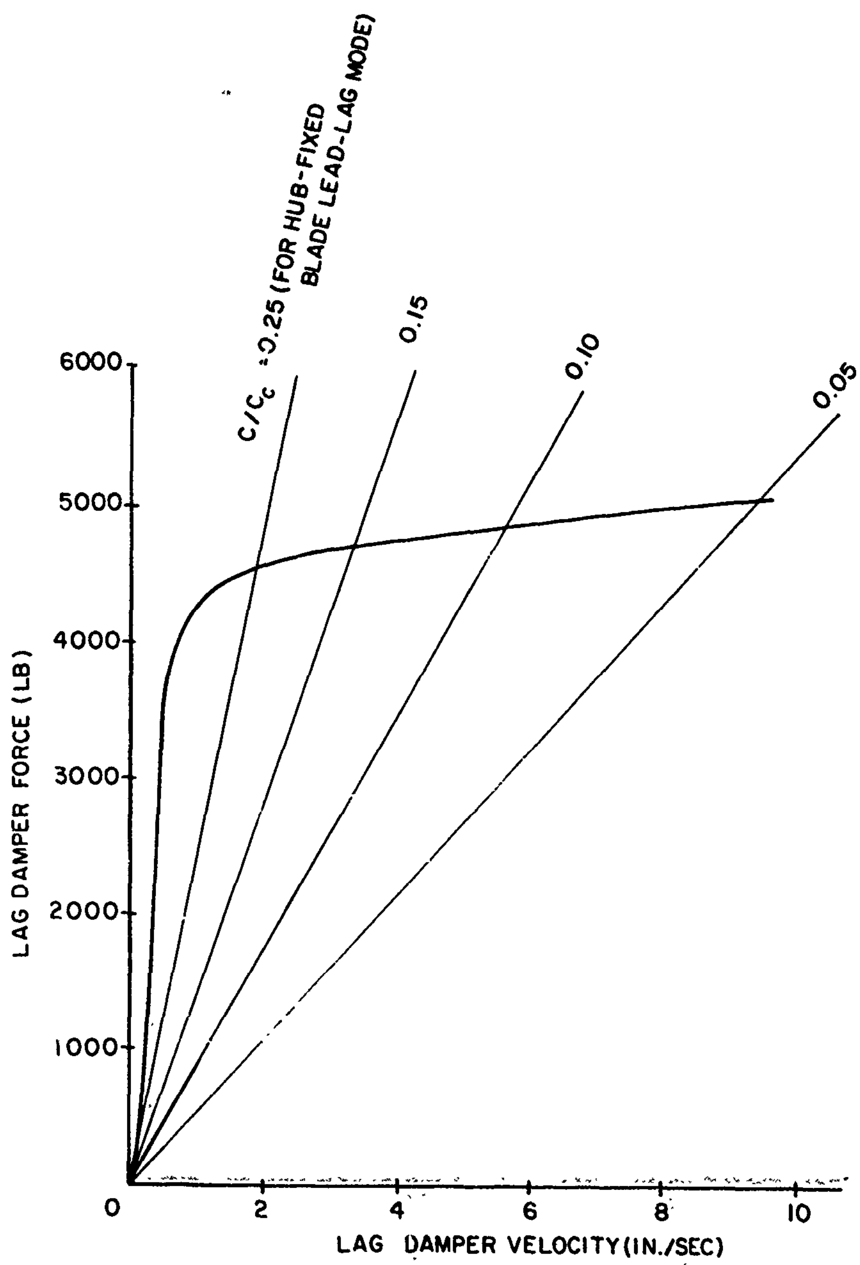

Figure 59. Lag Damper Instantaneous Force-Velocity Relationship.

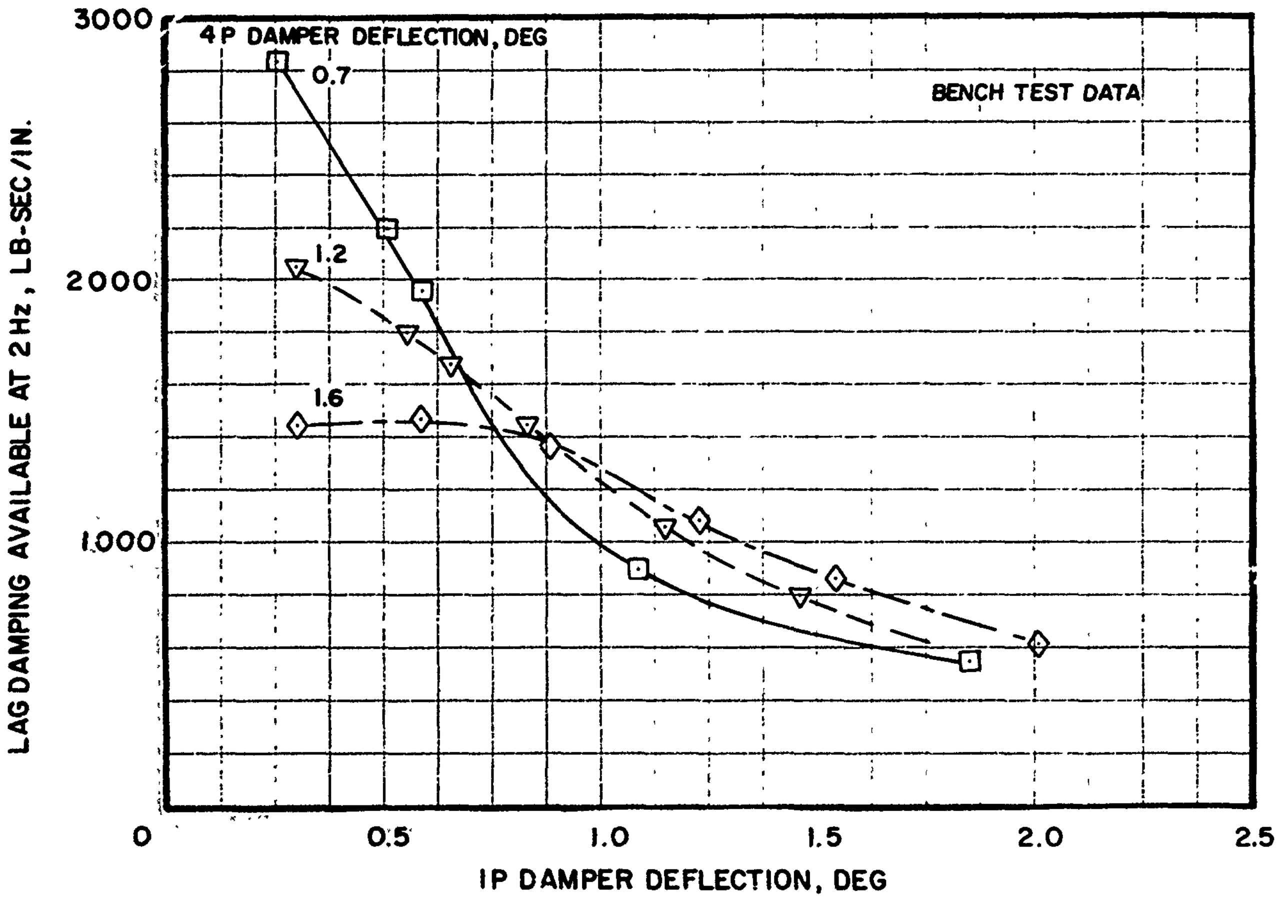

Figure 60. Lag Damping Available at 2 Hz vs Superimposed 1P and 4P Motion.

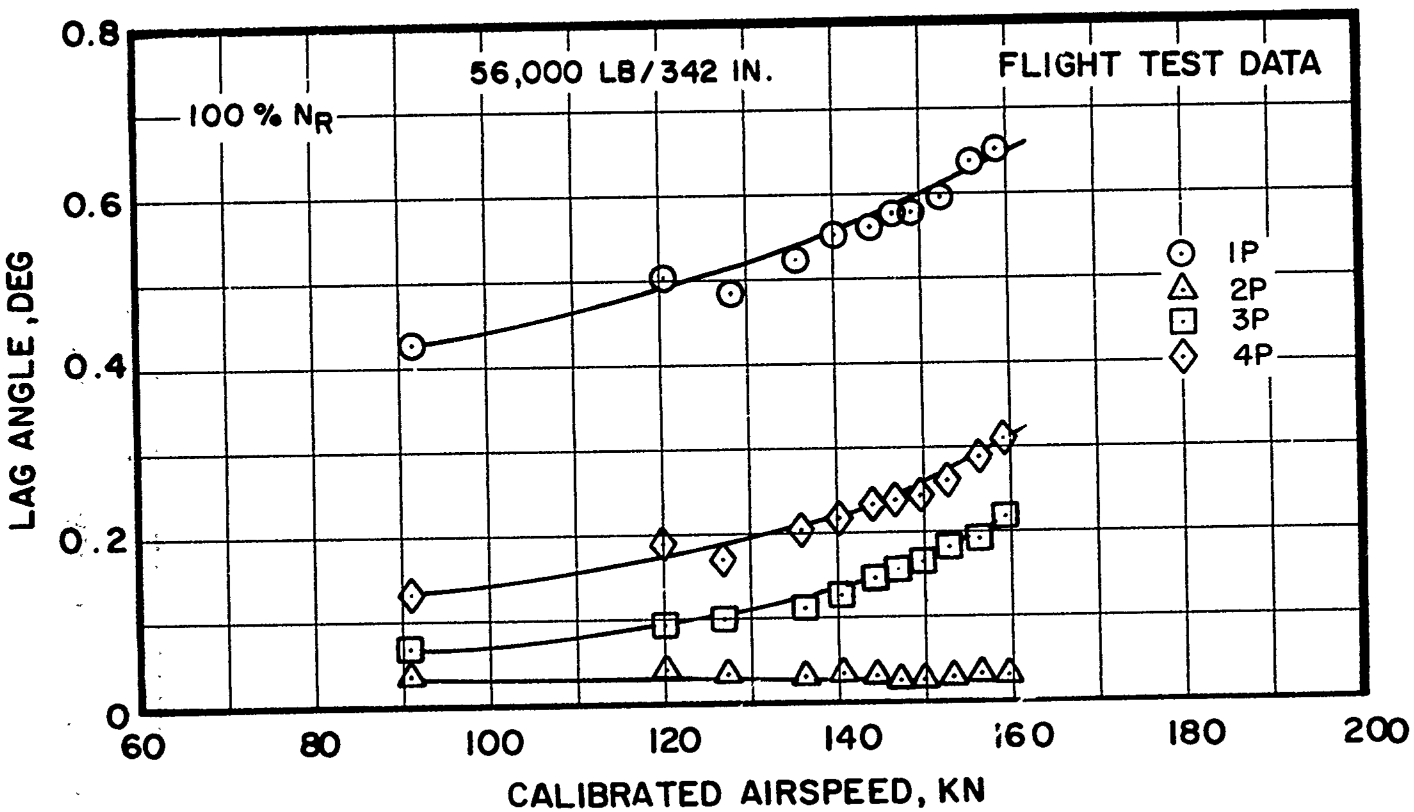

Figure 61. Lag Angle 1P → 4P Motions vs Calibrated Airspeed.

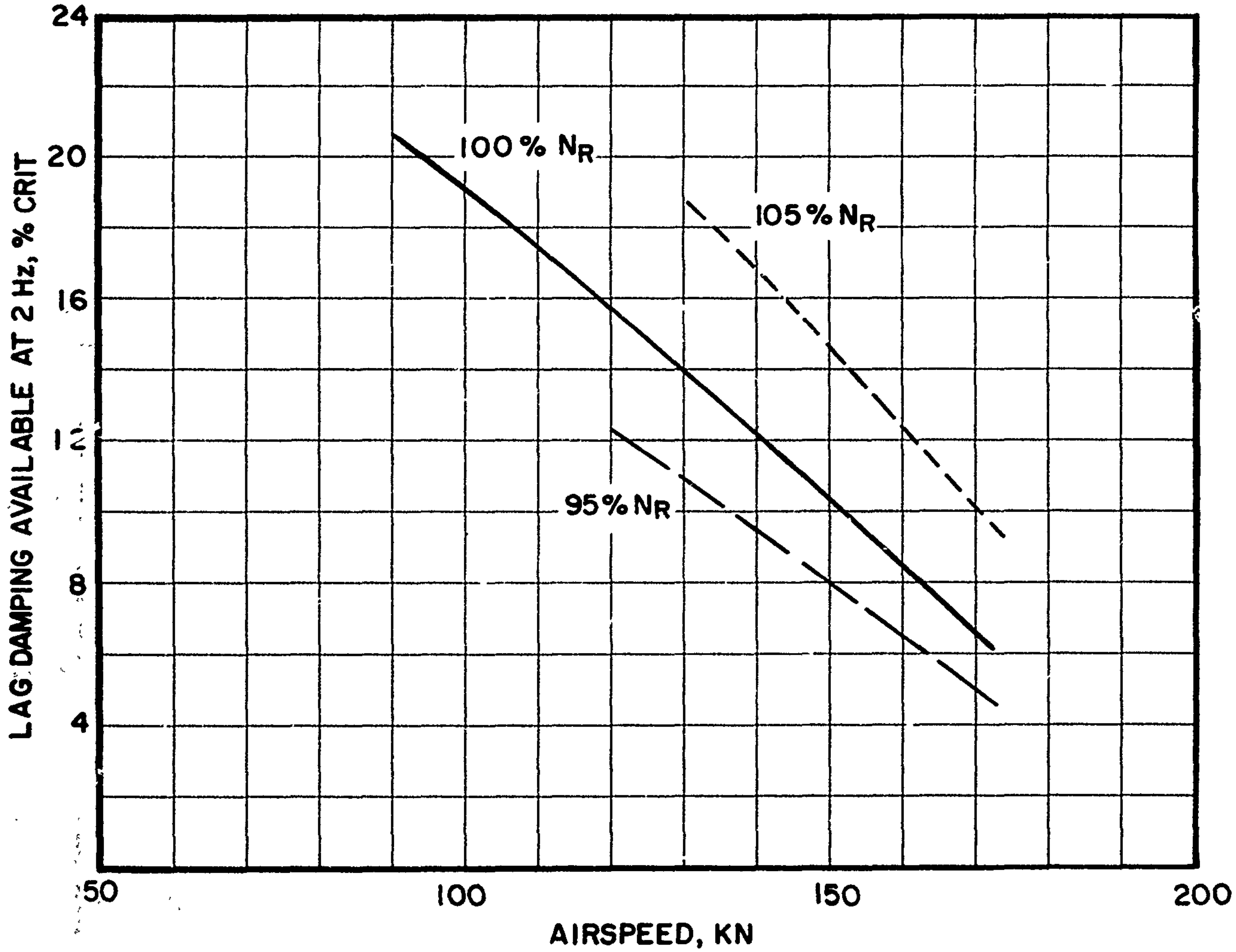

Figure 62. Lag Damping Available at 2 Hz vs Airspeed.

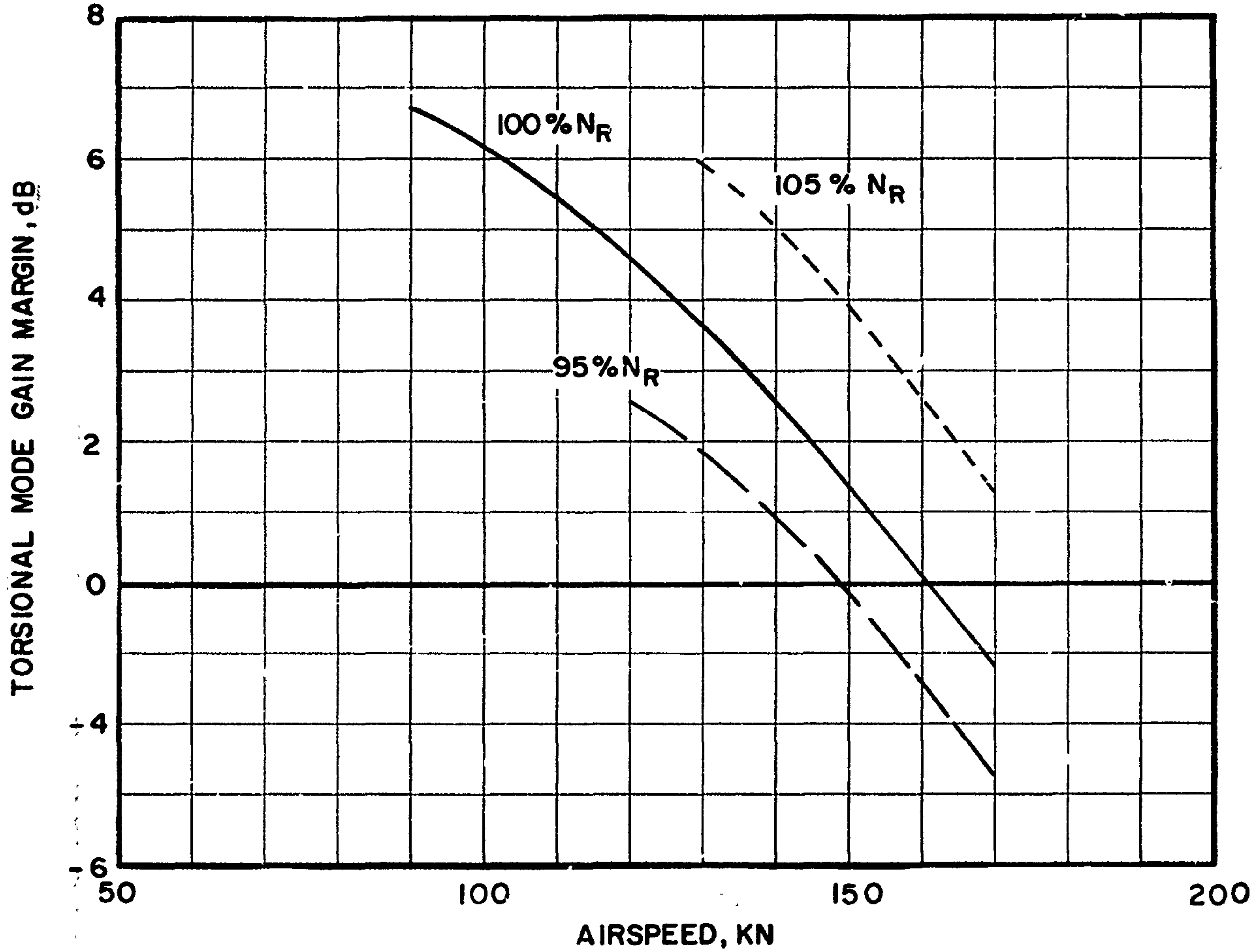

Figure 63. Torsional Mode Gain Margin vs Airspeed.

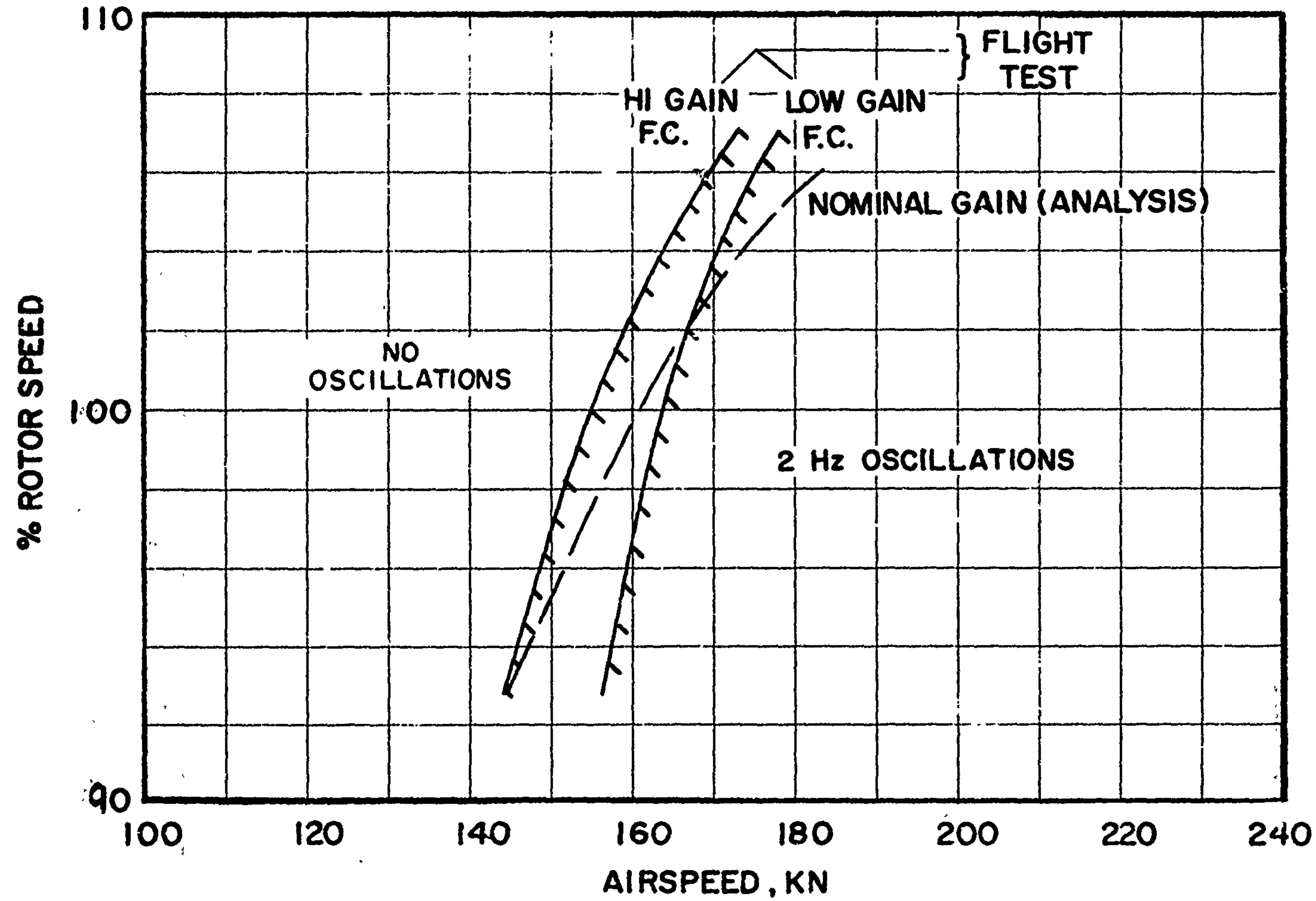

Figure 64. Oscillation Boundaries of Torsional Mode.

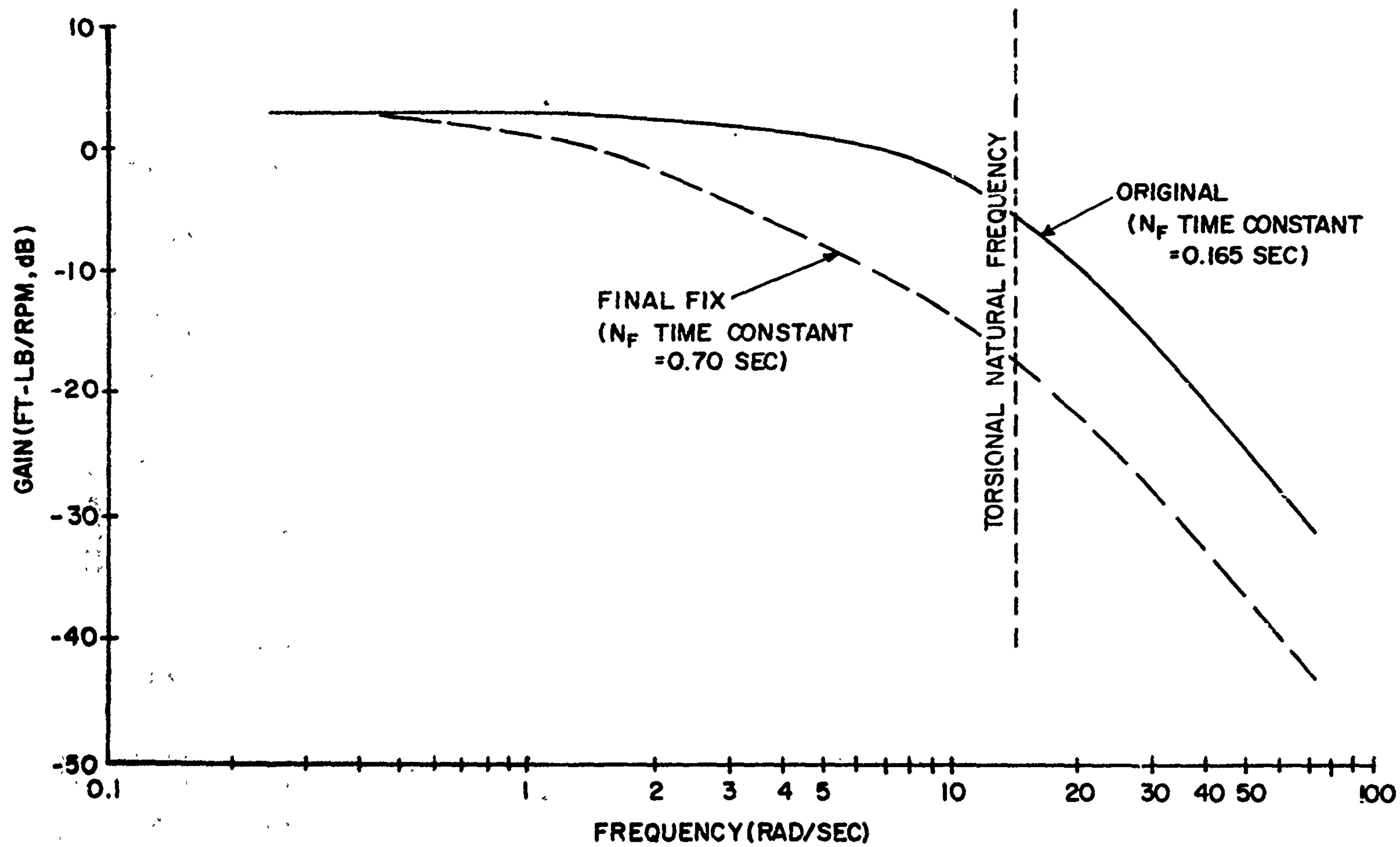

Figure 65. Frequency Response of Fuel Control System.

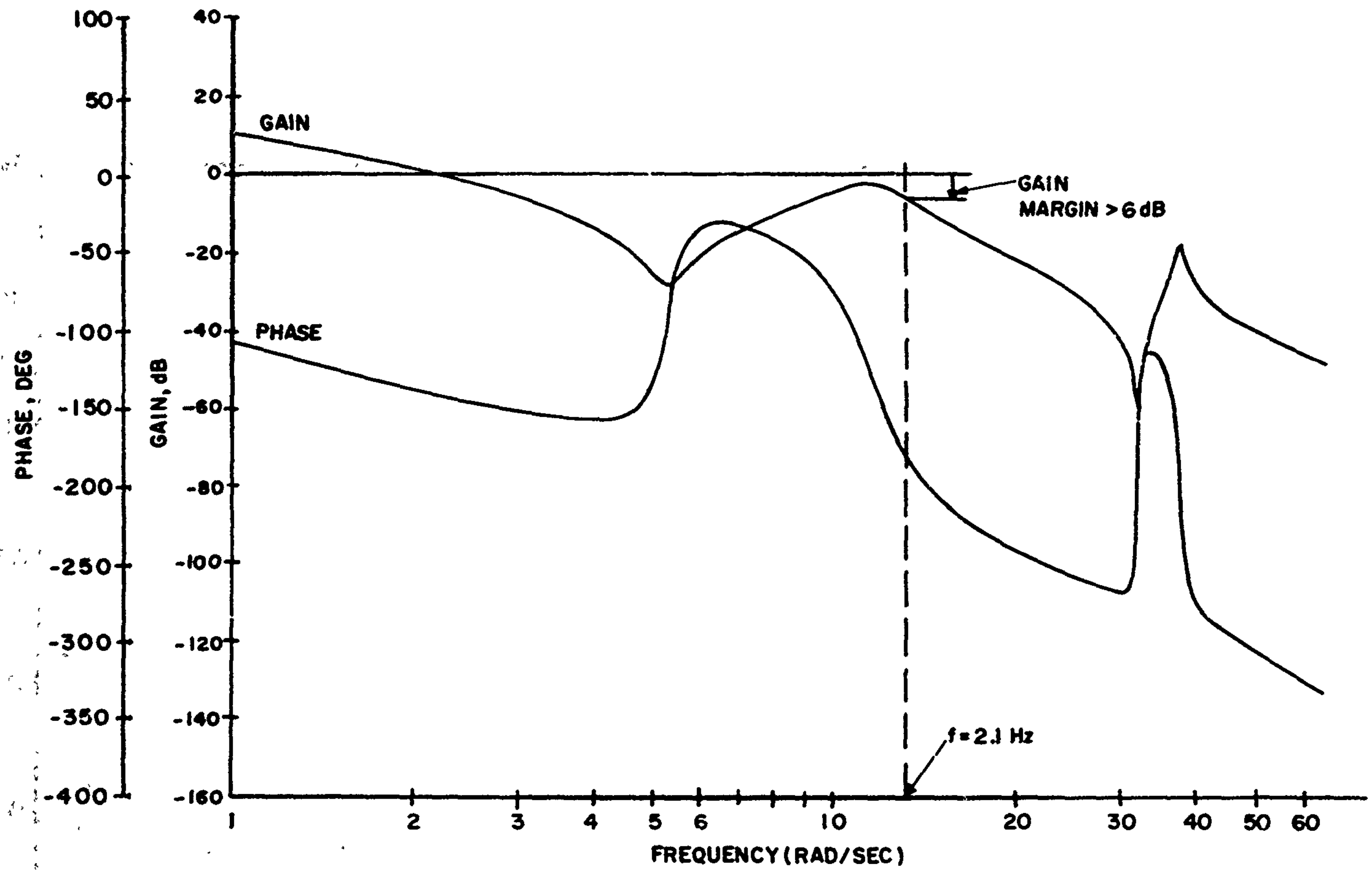

Figure 66. Torsional System Bode Plot, with Solution.

TRANSIENT RESPONSE PROBLEM

Problem 18. CH-54/JFTD12A-4A: Excessive Transient Droop

Description of Problem

During the course of testing conducted in 1968/1969 as a part of the U.S. Army CH54A and B heavy lift helicopter program, it was established that the JFTD12A-4A engine fuel control acceleration response characteristics prevented the engine from satisfying aircraft power requirements resulting from large collective pitch increase.

The fuel control characteristics were such that a 10% speed error, between the fuel control speed governor setting (N_2) and power turbine actual speed, was required before the fuel control would put the engine on its max acceleration line. As a result, in operations requiring a large collective pitch increase, such as a flareout after a fast let-down or during a jump takeoff, main rotor rpm would fall below 90% before engine power output maximized. At this point, tail rotor control power capability was at a minimum due to the reduced (80-90%) main/tail rotor RPM. Also at this point, engine output power (torque) was reaching its peak in an attempt to restore 100% power turbine/rotor RPM. As a result, the tail rotor was unable to develop the thrust required to counteract the now maximized engine input torque, and the resulting aircraft yaw would be adverse enough to threaten aircraft stability.

Investigation

The JFTD12A-4 engine acceleration characteristics, as substantiated in engine test cell operation, were excellent. Starting from a flight idle condition, the engine reached 95% of military power in less than 2 seconds. When installed in the aircraft, however, the 10% speed error required by the fuel control effectively nullified the engine's excellent inherent acceleration capability.

Solution

The solution was a modified fuel control which was incorporated into the JFTD12A-5A engine which replaced the JFTD12A-4A. The modified fuel control system would put the engine on its maximum acceleration schedule as soon as a 3% speed error occurred rather than the original 10%.

CONCLUSIONS

This survey of engine/airframe/drive train interface dynamics problems encountered at Sikorsky over the past 20 years reveals some definite trends.

Problems in the forced vibration class have regularly surfaced at approximately a constant rate over the last two decades. This would suggest that these are a function of the designs of the particular installations and could be further minimized by continuing to improve design criteria and design approaches, particularly for growth systems.

Problems in the self-excited class have surfaced in recent years at an increasing rate. These phenomena involve more components of the rotor/airframe/drive system and are probably due in large part to the increasing demand for highly responsive systems. In future years, the satisfactory design of the fully integrated system will require more coordination among the airframe, engine and fuel control manufacturers. Success is also expected to require an interdisciplinary design approach, involving propulsion, dynamics, and handling qualities, within the airframe manufacturers organization.

RECOMMENDATIONS

To reduce the rate of occurrence of forced vibration problems, it is recommended that:

1. Generalized design criteria be formulated which specifically address past problems.

2. Good design and development approaches be formulated involving component and multiple component substantiating tests to minimize risk.

The recent increase in self-excited vibration problems involving the rotor/drive train/engine/fuel-control system points to a need for increased understanding in this area. Improvements are required in the analytical methods of prediction to enable these often complex problems to be foreseen before hardware is built and solutions become more costly. Design compromises, such as those between engine response and torsional stability requirements, must be thoroughly understood early in the design cycle to arrive at optimal designs.

Specific recommendations are:

1. Develop analyses which model the fully coupled rotor/drive train/engine/fuel control/fuselage system.

2. Conduct parametric analyses to define component interactions.

3. Correlate with past experience.

4. Based on these results, formulate interdisciplinary design criteria and design guidelines.

5. Formulate an integrated design approach for stability, transient response, handling qualities and installation considerations.

6. Use results to define requirements for advanced subsystems, in particular an electronic fuel control.

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